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Heating Piping & Air Conditioning

DESIGN
INSTALLATION
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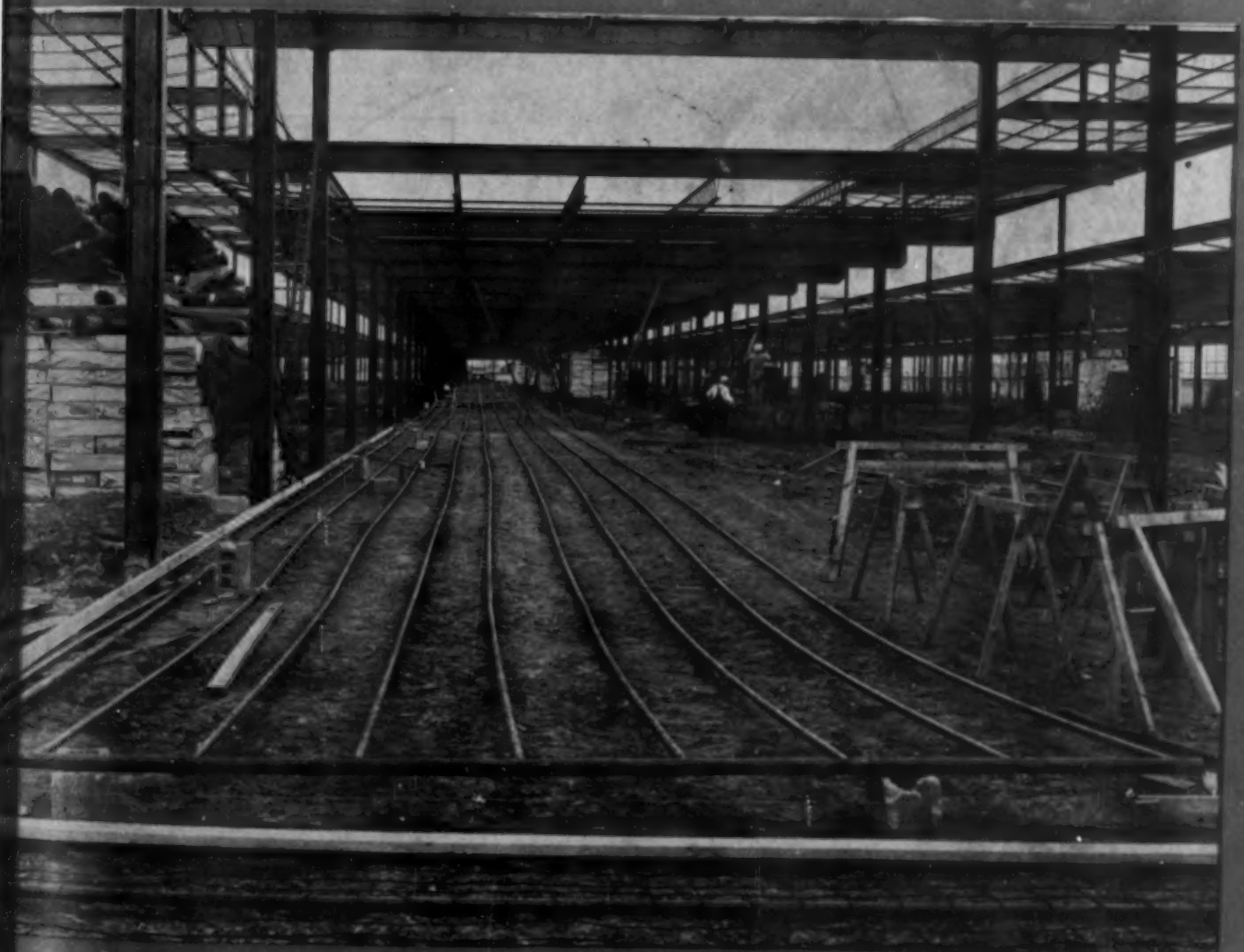
MAINTENANCE

SEPTEMBER ★ 1946

INDUSTRIAL
COMMERCIAL
INSTITUTIONAL AND
PUBLIC BUILDINGS

105 — *Journal Section of the American Society of Heating and Ventilating Engineers*



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Gas fired water heaters—120 of 'em—supply radiant space heating coils at big new plant *—see page 69*

Trouble-Proof

AGAINST CORROSION, CLOGGING AND PRODUCT CONTAMINATION

GRINNELL-SAUNDERS DIAPHRAGM VALVES

Screwed or Flanged Patterns

Users of compressed air, gases, viscous or corrosive liquids, food products and fluids containing suspended solids – this is the valve that may eliminate your troubles!

- ★ No packing glands to demand constant attention.
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- ★ The large area of contact of the diaphragm on the seat, plus the resilience of the diaphragm, permits positive closure even when foreign matter is trapped.
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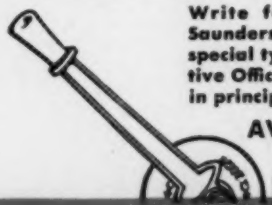

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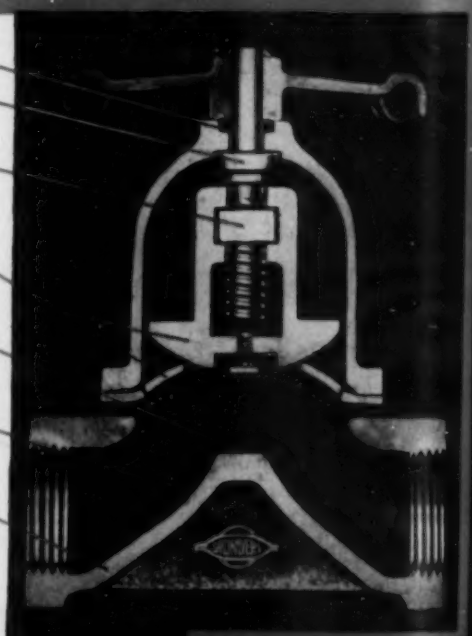
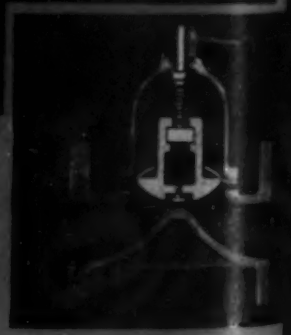
The "quick operating" lever design is one of several types.



VALVE OPEN



VALVE CLOSED

GRINNELL

FOUNDED 1850

WHENEVER **PIPING** IS INVOLVED

THE EDITOR'S PAGES

NBFU STRESSES NEED FOR OIL BURNER CHECKUP

Fires involving oil burners are increasing due to conditions brought about by the war, engineers of the National Board of Fire Underwriters, 85 John St., New York 7, N. Y., said last month. Principal cause is lack of proper maintenance, which has been due to lack of materials and the scarcity and high cost of trained mechanics.

Many burners installed during the war do not meet the minimum standards for safety and reliability. Others are just now wearing out after years of service, having been used longer than the reasonable life of such devices.

"Any mechanical device, as it nears the end of its usefulness, becomes more difficult to maintain properly, and requires good maintenance to a greater degree," the national board said. The board hopes that increased production of oil burners will soon correct the hazard of the wornout burner.

Chief among the faults causing oil burner fires is simple failure to check and clean it once a year.

"Continued operation of the burner without cleaning and removal of carbon deposits inevitably results in clogging of the burner orifice," the engineers said. When this happens, the flame goes out. Pressure builds up behind the obstruction. Frequently this pressure blows out the carbon, and oil again flows into the heated combustion chamber. So much oil is vaporized that combustion is incomplete. The result is clouds of sooty smoke puffing out of the furnace.

When a furnace explodes, that can also be blamed on failure to keep the burner clean. Under certain conditions, when pressure blows the obstruction out of a clogged up orifice, the vapor creates a mixture not proper for combustion. This delays the ignition so that when it does occur it comes violently, sometimes blowing off the furnace door and shooting out flame.

The board pointed out that oil burners which are neglected not only are unsafe but are uneconomical. When combustion is incomplete, gases which should create heat are unburned and are carried up the chimney. This results in heavy accumulations of burnable soot in the chimney. Oil soot discharged from chimneys also lodges on roofs and sides of buildings. This oily deposit must be removed before buildings are painted, thus increasing cost of painting.

Another source of current trouble is the oil burner that was removed for lack of oil during the war and later re-installed without the aid or supervision of a competent oil burner service mechanic.

A study of fires in which oil burners are involved shows that two facts are of prime importance, the board's engineers stated. First, the installation must be properly made by a well-trained man following the manufacturers' instructions, and second, the installation must be properly serviced once a year. A thorough check on the automatic control device and proper oiling and cleaning of the motor and other moving parts should be included in any checkup.

REVIEWS DEVELOPMENT OF PIPING COLOR CODE

The historical growth of the American Standards Association color standard was reviewed, on the anniversary of its approval, in an article by Henry G. Lamb, ASA safety engineer, published recently in Industrial Standardization, the ASA's journal. The following paragraphs on identifying piping systems have been taken from this discussion:

In 1928, an American Recommended Practice Scheme for the Identification of Piping Systems was approved. This extended the previous use of colors in industry to the identification of various piping systems used for steam, water, liquids, and air throughout a plant. Here, the color red was extended to include the identification of sprinkler piping as part of the fire protection system. The range of colors from yellow to orange was recommended to designate piping containing dangerous materials such as acids, caustics, and high pressure steam. Green was recommended for safe materials such as ordinary drinking water, low pressure air, and other safe liquids; blue was designated for protective materials other than fire protection; and purple was to be used for the identification of extra valuable materials.

This scheme had as its primary purpose prevention of accidents to pipefitters and maintenance men who, from time to time, are called upon to cut into a piping system either to make an extension or to make repairs. If all of the pipes are painted the same color and several lines are running parallel and in close proximity to each other, it is sometimes difficult to trace out a line and ascertain that it is a harmless water line that is being cut into, rather than another pipe which may contain a corrosive acid.

MARKET STUDIES BEGUN OF REFRIGERATION, CONDITIONING

Market studies of important industries to determine the potential requirements for refrigeration and air conditioning equipment will be undertaken by the Air Conditioning and Refrigerating Machinery Association, 717 Southern Bldg., Washington, D. C., representing many of the larger manufacturers of air conditioning and refrigerating equipment.

Announcement of the establishment of the research and statistical department to conduct this work was made by William B. Henderson, executive vice president of the association. The object of the department will be to maintain current information on the rapidly expanding use of refrigeration and air conditioning machinery and to conduct special studies by assembling, collating, and analyzing all available data bearing on the use and the probable new demands for refrigeration and air conditioning machinery.

Early attention will be given by the new department to a survey of the food industry, one of the largest users of refrigerating and air conditioning equipment. The first phase of this work, already under way, deals with the dairy industry and its sales potential for refrigeration equipment. A part of this study

THE EDITOR'S PAGES

dealing with dairy farming already has been completed. The work will be carried on to supplement market research activities of member companies for advertising guidance, and to provide a source of public information about the increasing use of refrigeration and air conditioning equipment for production purposes by other industries. The industry has lacked statistical data, especially relating to the use of air conditioning, due to its rapid growth in the period just prior to the war.

REMA ADDS 100TH MEMBER

At the midsummer meeting last month of the directors of the Refrigeration Equipment Manufacturers Association, it was announced by H. F. Spoehrer, president, that membership in the organization had passed the 100-mark.

The 100th company voted into the membership at the meeting is Drayer-Hanson, Inc., manufacturers of heat transfer equipment. During the last three years, REMA has more than doubled in size.

Mr. Spoehrer credited the growth of the association to its various overall and product section activities and to its cooperation with other associations in the industry.

SPONSORS NOMENCLATURE STANDARDIZATION

In connection with the effort to standardize abbreviations, graphical, and letter symbols used by the air conditioning and refrigeration industry, the American Standards Association nomenclature committee B53, sponsored by the American Society of Refrigerating Engineers, is endeavoring to find out just what is now being used by the industry.

It was felt by the committee personnel that the best way to achieve standardization of the various abbreviations and symbols was to obtain from the industry samples of what are now being used, carefully study and compare them, and then arrive at a decision as to best ones to adopt as standards.

Accordingly, the committee is asking representative firms engaged in refrigeration and air conditioning for the following information:

1) What graphical symbols do you use in representing refrigeration equipment within the refrigeration cycle?

2) What letter symbols do you use?

3) What abbreviations do you use?

The scope of the inquiry is limited mainly to nomenclature used in the refrigeration cycle because of the very complete sets of standards already adopted for allied industries and professions such as electrical, mechanical, hydraulics, and piping. Appropriate selections of these standards will be included in the final report of the committee.

The committee hopes that any firm or individual that is interested in this matter will send in a set of the symbols and abbreviations that they use, together with any suggestions they may care to offer. Communications should be addressed to D. F. Hayes, Secretary, Nomenclature Committee B53, American Standards Association, 70 E. 45th St., New York 17, N. Y.

SEES SUFFICIENT STOKER COAL

The supply of stoker coal for use in residential and apartment house stokers apparently will be sufficient to meet most or all requirements of present users and prospective users during the next heating season, said the National Coal Association last month, as the result of a preliminary survey and spot-check made in a number of major bituminous coal producing districts, which supply a considerable portion of the domestic stoker coal sizes for the major consuming areas in the eastern, middlewestern, and southern states.

This survey was made as a result of meetings and investigations by representatives of the bituminous coal industry and the stoker manufacturing industry. The coal group consists of representatives of the sales advisory committee of Bituminous Coal Research, Inc., and members of the board of directors of the Stoker Manufacturers Association.

The report further disclosed that in some major producing districts an increase in the production of double screened domestic stoker coal may be expected in the period beginning August 1, 1946, and ending March 31, 1947, as compared with the same production period a year ago. It also indicated that an increase in the production of 1¼ in. and 1½ in. coal for use in commercial and industrial stokers is anticipated in some of the major producing districts.

DRAFT DEFERMENTS FOR HOUSING WORKMEN

The machinery by which skilled workmen may be recommended for draft deferments in order that they can continue to work in the veterans emergency housing program has been set up by the National Housing Agency and the Selective Service System.

NHA will assist the director of selective service in recommending the skilled construction workers who should be deferred from military service. The following types of construction workers have been determined by the director as eligible for certification by NHA:

1) Skilled workmen engaged in home building where shortages of such men have been established. Such persons must have had at least three years of training and experience in key building crafts, such as bricklaying, plastering, plumbing, and carpentering. It must be established that the loss of such persons is delaying or immediately threatens to delay the veterans housing program.

2) Managerial or supervisory personnel possessing a minimum of three years' experience in home construction.

Employers seeking deferment for a given individual, or self-employed construction workers, must file selective service form 42-A (special-revised) with the appropriate NHA regional office or NHA locality expediter. The application for deferment will be investigated in the field. Such investigation will include consultations with the U. S. Employment Service and local union officials whenever appropriate.

Heating Piping & Air Conditioning

with which is merged the Aerologist

Design, Installation, Operation, and Maintenance: Industrial, Commercial, Institutional, and Public Buildings

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Associate Editor
Thomas V. Johnson

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Wallace J. Osborn
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Telephone: Murray Hill 9-8293

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George C. Cutler
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Telephone State 6916

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Publisher
Frank P. Keeney

Manager
Chas. E. Price

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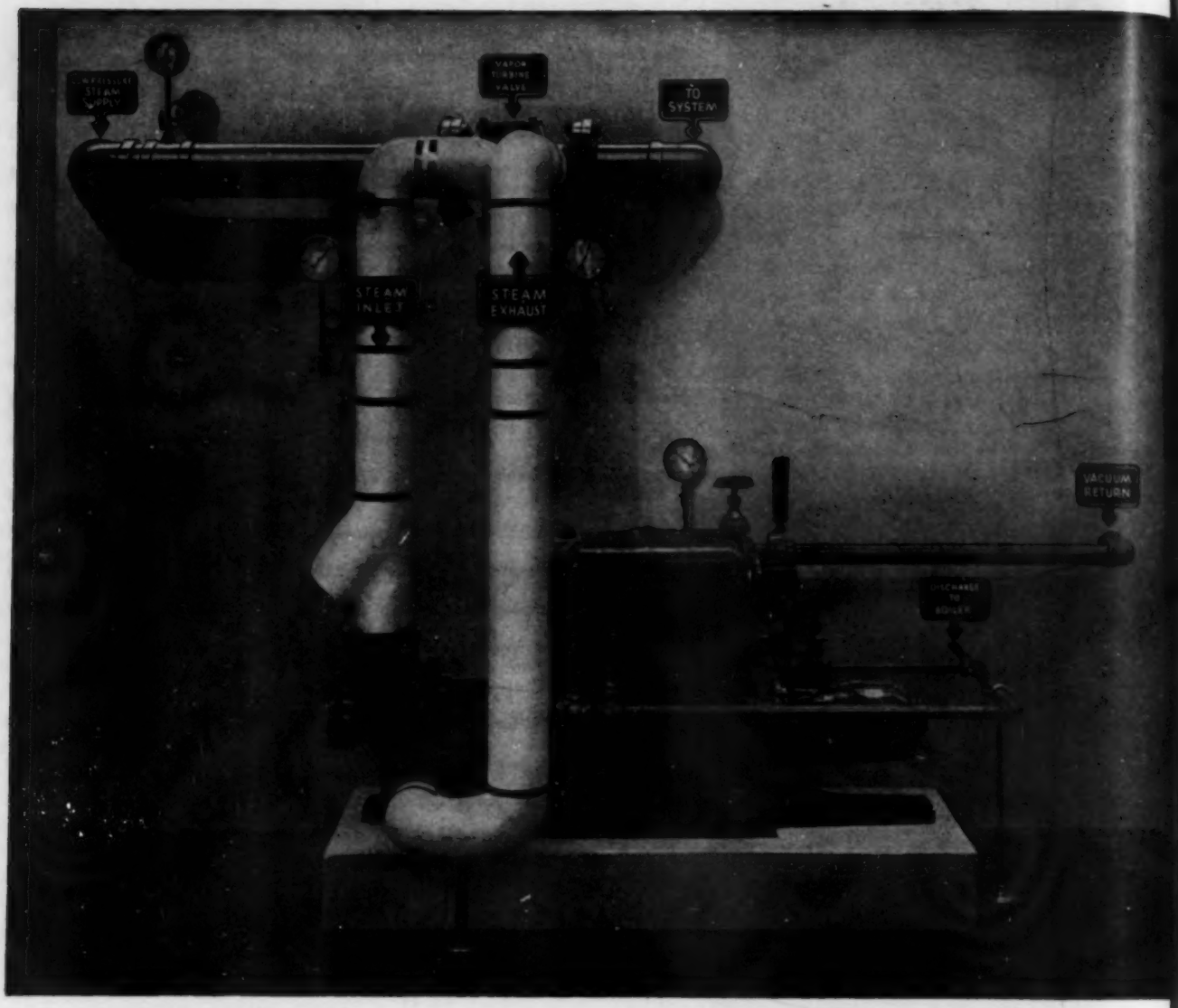
Interrupted Electric Service wil not stop this NASH Heating Pump



In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is independent of electric current failure, and continues to operate as long as there is steam in the system.

This is because the prime motive power of this economical pump is a special steam turbine, controlled by a unique "Vapor Turbine Valve", which automatically bypasses from the heating main a small portion of steam, the exact amount necessary to develop the power needed to remove the condensate and maintain the required vacuum on the system. Even this small amount is passed immediately back to the mains, and goes on to the system with little heat loss. This pump operates on any system, high or low pressure.

The Vapor Turbine is a most economical pump, for the elimination of electric current does away with current cost, the largest single item in the operation of an ordinary return line heating pump. Bulletin on request.

THE NASH ENGINEERING COMPANY
205-A WILSON ROAD, SOUTH NORWALK, CONNECTICUT, U. S. A.

UNUSUAL JOB

120 Water Heaters Meet Space Heating Needs of Big Radiant Installation

New A. O. Smith Plant Has Unit System, With Gas Fired Heaters Supplying Individual Floor and Ceiling Coils

AN UNUSUAL experiment in plant heating is being made at the Kankakee, Ill., works of the A. O. Smith Corp., where 120 volume water heaters, operating on natural gas, will furnish hot water to floor and ceiling radiant heating coils to meet the entire space heating needs of this new factory. According to C. W. Colby, designer of the plant's heating system and the engineer in charge of its installation, the novel scheme will show economies both in initial and operating costs. The savings possible with such a "unit" system in matching the heat generated with the heat requirements (as determined by the outside temperature) are not usually fully appreciated, he says; with a large number of separate systems, only those needed at any particular time need be supplying heat. With a number of small, automatic units, according to Mr. Colby, operation of the heating system is simplified, and the capital investment in a boiler house, a stack, etc., is saved. Another consideration was that the use of water heaters for plant heating will give the owner, who distributes this heater, valuable experience in "living with it" under actual operation.

Similar to Mount Holyoke

Mr. Colby was involved, some 12 years ago, in applying a somewhat similar unit scheme at Mount Holyoke College, South Hadley, Mass., where an existing central boiler plant was replaced by 120 individual oil-fired burner-boiler units, all situated in one boiler room. Worthwhile economies in both first and operating costs were achieved in that case, he states. (A description of the Mount Holyoke job was published in the November 1934 HPAC, and a report on its performance appeared in the August 1938 HPAC.)

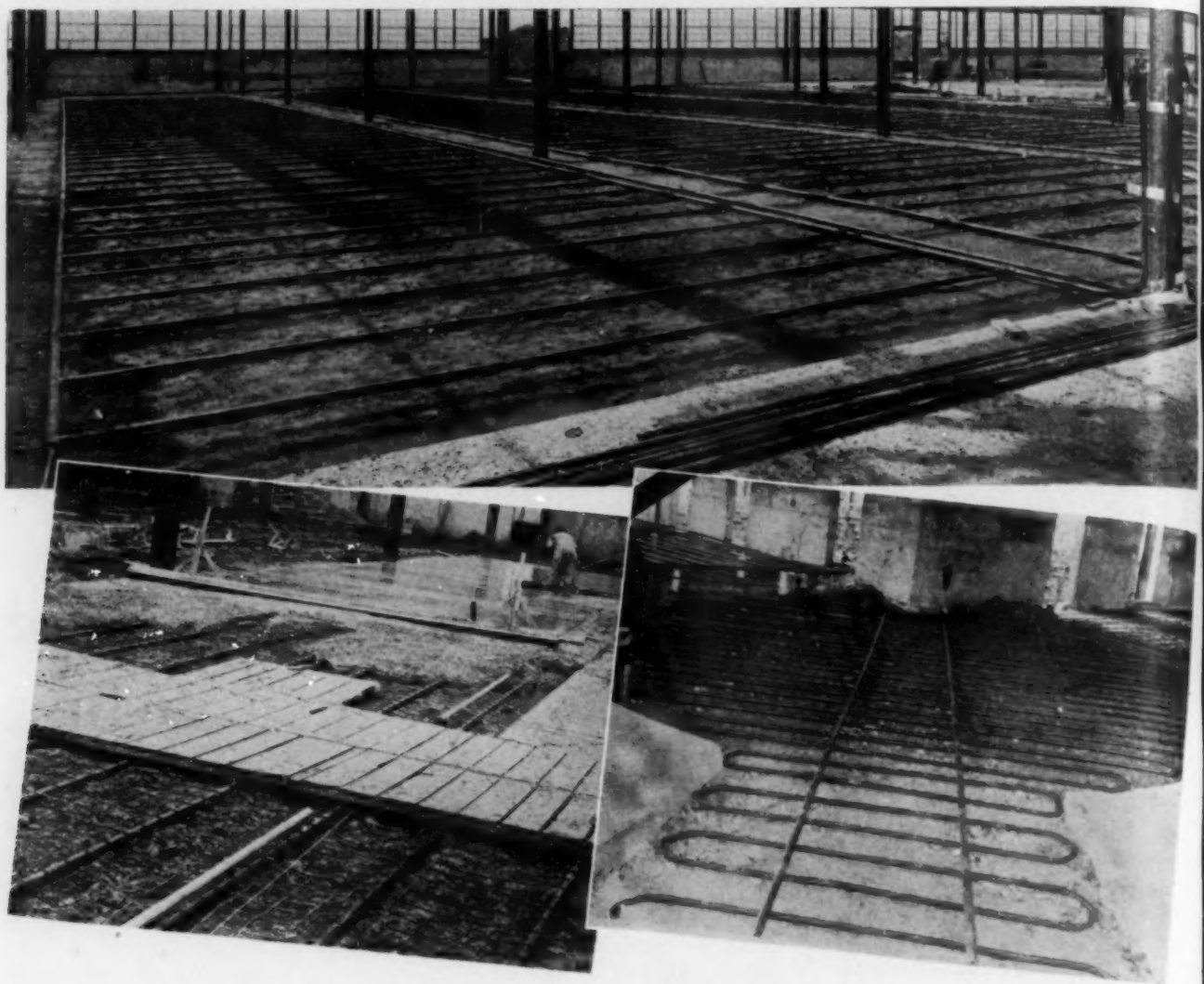
At the Kankakee works will be manufactured all of the company's domestic storage water heaters, both gas and electric, with production at the rate of 250,000 heaters per year.

The plant is largely of one-story construction, 1200 ft long and 315 ft wide. In addition, there are mezzanine floors for laboratories, comprising 10,650 sq ft, and office space consisting of 22,788 sq ft. The total is 415,188 sq ft.

The radiant heating system in this plant is one of the largest on record. It consists of 40 miles of piping and will circulate 180,000 gallons of water an hour.

The piping consists of wrought iron, copper, and steel. The wrought iron ranges from 2 in. down to ¾ in., the copper from 1 in. to ⅜ in., and the steel is ⅝ in. copper brazed steel tubing. The use of these materials was dictated primarily by engineering logic. Secondly, however, necessity and exigency played their part, and in the third place the opportunity to experiment with various materials was not overlooked.

AT THE NEW A. O. Smith plant nearing completion at Kankakee, Illinois, for the manufacture of 250,000 domestic storage water heaters a year, volume water heaters operating on natural gas are being used to meet all of the space heating requirements. One hundred and twenty of these units serve as many floor and ceiling coil installations, under the control of 10 outside master thermostats which will govern the temperature of the water circulated in accordance with outdoor temperature. . . . In addition to heating the plant, the installation will serve as a testing ground to answer a number of radiant heating questions.



Typical grid construction in wrought iron pipe. It consists of long 2 in. headers, approximately 140 ft, with short 1¼ in. laterals, approximately 20 ft, with varying center to center spacings determined by the rate of heat loss

Wrought iron pipe in grids—1¼ in. laterals on 24 in. centers from 2 in. headers—placed directly on the dirt grade with wire mesh and concrete superimposed

Entrance lobby, south office building, showing the special wrought iron grid to apply to the circular construction of the space involved

The plant is divided into 48 bays, each of which runs clear across the 315 ft width of the plant and is 25 ft wide. Two water heaters serve each bay, each heater supplying panels which cover half a bay. The first step was to construct a concrete roadway down the center of the plant to allow trucks to bring in supplies.

The Plant Proper

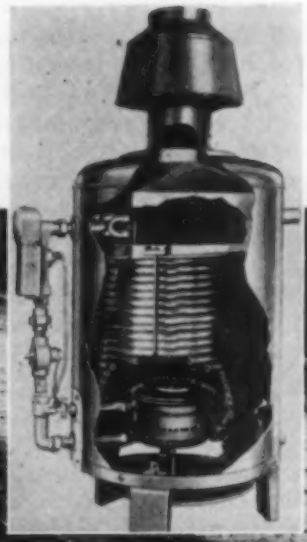
Eighty per cent of the installation in the floor of the plant is of grid type construction, consisting of headers and laterals of wrought iron pipe. Some of the grids have long 2 in. headers (140 ft) and short 1¼ in. laterals (20 ft), while others consist of 20 ft headers and 140 ft laterals. Center-to-center dimensions range from 4 ft to 8 in.

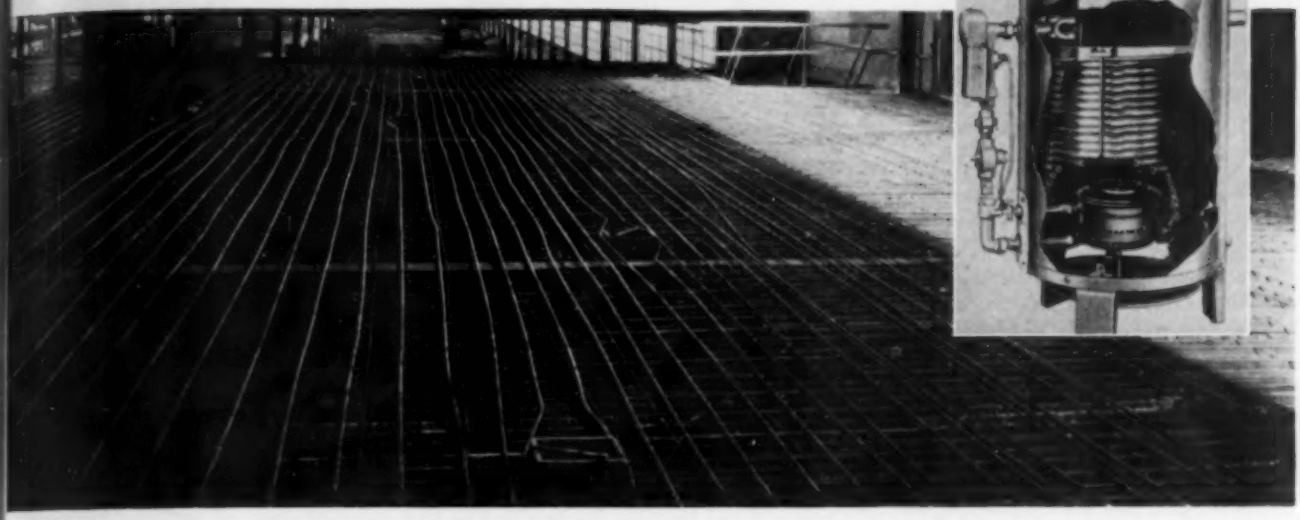
There are obvious advantages to the grid type construction. It is the best type to meet possible future changes in floor arrangements and foundations because new headers can be installed as needed around the new foundations. It permits mass production of headers and laterals and makes possible many welds at the point of manufacture rather than on the job. It also permits the use of random lengths of pipe, which is a considerable economy. Arc welding was used for joining laterals to headers, and gas butt welds were used in the laterals.

Nearly all of the rest of the plant floor has been covered with a combination of grid type and continuous coil construction. Necessity was the reason for much of it. The company was able to procure pipe suitable for continuous coil construction but not for the grid type laterals. Headers were introduced into the continuous coil construction for two principal reasons: (1) To reduce resistance to flow where it was too great for the capacity of the pumps, and (2) to guard against future changes in floor arrangement and foundations. The effect of using headers in a continuous coil panel is to break it up into two or three smaller continuous coil circuits.

In one area a single continuous coil was used for a purely experimental reason. It is near a large process furnace in which the temperature will exceed 1600 F, and heat will flow from this furnace to the pipe coil. By circulating the water

This is the water heater used, 120 of which will furnish hot water for the radiant heating system. A bypass hookup has been used so that the full capacity of the heater can be utilized. Water enters the unit at the top and flows downward through the tubing which is coiled in such fashion that it becomes, in effect, the wall of the combustion chamber. The heat exchanger at the top exposes 30 sq ft of transfer area to a total water content of four pints





Copper brazed steel tubing in loops approximately 100 ft long in the 3 in. thick roof decking and concrete floor construction of the appliance testing laboratory, mezzanine floor, southeast corner of plant

without otherwise heating it, the rate of heat transfer through the concrete slab will be determined.

Mezzanine Floors and Office Areas

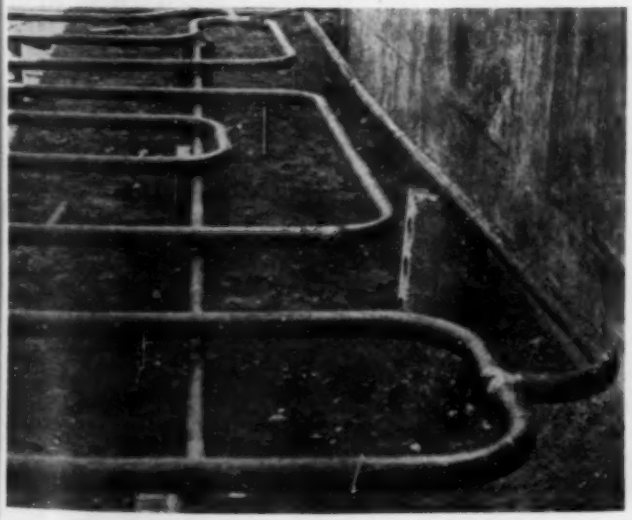
All of the heating installation in the mezzanine floors is of copper brazed steel tubing, as it was the only tubing available that would fit into the reduced thickness of the concrete slab that forms the floor. The tubing is ⅝ in. outer diameter with a wall thickness of 0.049 in. It is of grid type construction throughout, with center-to-center distances varying from 6 to 12 in., depending upon the estimated rate of heat loss.

For the first floor offices, wrought iron was used in grid type construction. Then, in the ceiling of the first floor offices, an auxiliary wrought iron surface was used at the outside exposures to add radiating surface for the second floor offices at the windows, and for the ceiling of the first floor. The office area heating job was completed with the installation of copper in a grid type and continuous coil construction, in the ceiling of the second floor offices so that heat will radiate downward.

The Water Heaters

The water heaters furnishing hot water to the radiant heating coils are, in general, situated along the monitor—requiring no floor space—with supply and return lines running down the columns to the coils. In two cases, because of the building design and layout, groups of the units are installed in heater rooms. Even in the case of the heater rooms, however, each heater supplies its own separate radiant heating coil.

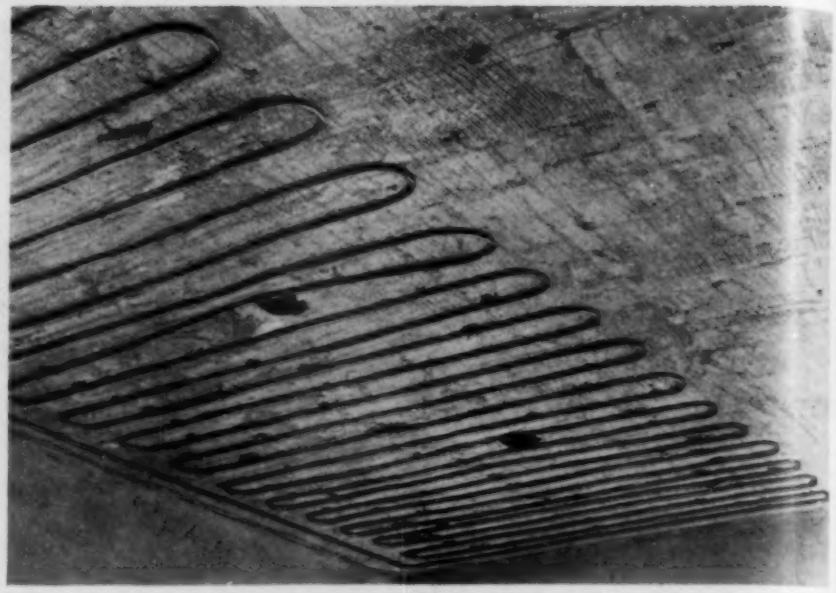
Two inch wrought iron pipe used in a combination of grid type construction and continuous coil construction, with concentration of surface at the outside wall



Copper brazed steel tubing at the ceiling (embedded in the plaster) as an added concentration of heat at the glass exposure in the entrance lobby, south office building

The heaters have a rating of 195,000 Btu input on natural gas, and an AGA hot water rating of 1040 sq ft. The cold water enters at the top of the heater, and flows downward through tubing coiled in such fashion that it becomes, in effect, the wall of the combustion chamber. Each heater is vented directly through the roof, the ones in the two heater rooms having gravity combustion air supply ducts and natural vents.

The heaters will furnish water at 140 F to maintain an estimated floor slab temperature of 85 F, and an air temperature in the plant of 65 F. These temperatures are based on an outdoor design value of –10 F. One study it is planned to make is on the question of maximum floor slab temperatures permissible with radiant heating installations. Mr. Colby feels that perhaps the usually accepted maximum of 85 F is too low, and that a 100 F floor slab temperature may be satisfactory if the occupants are walking about.



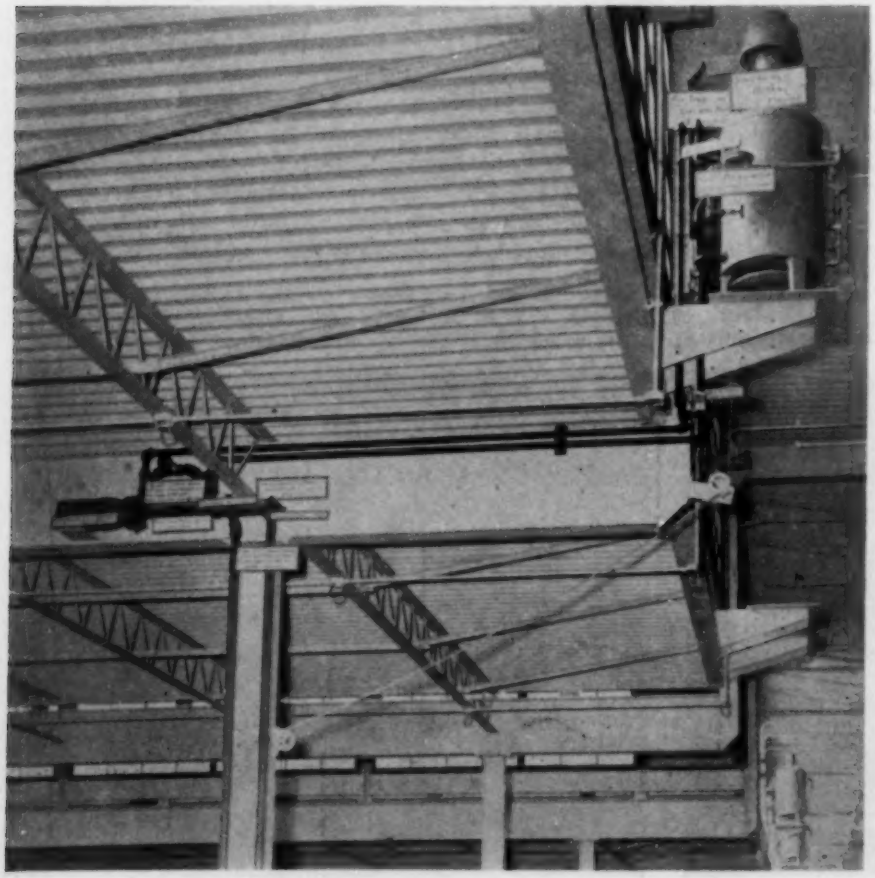
Three-eighths inch (nominal) copper tubing on second floor, south office building ceiling, installed on the "scratch" coat of plaster. Over this coat is placed approximately a ¼ in. brown coat and then about ½ in. of acoustical plaster. The tubes are ½ in. outside diameter and are installed on 4 in. centers at the outside exposure, changing to 5 in. centers, then 6 in., then 8 in., then 10 in. as the required concentration of surface decreases

Ten Outside Master Controls

The temperature of the water circulated in the floor and ceiling coils is governed according to the outdoor temperature by means of 10 outside master thermostats—two on the east exposure, two on the west, one each on the north and south, and two for each of the two office sections. Each of these 10 outside thermostats governs 10 to 15 inside submaster thermostats. The ratio maintained between the outdoor temperature and the water temperature in any one of the heating coils can be individually set at each submaster control, in order that the correct water temperature for each section of the plant for varying outdoor temperatures—as determined by local conditions, heat losses, process heat gains, etc., in each of the plant areas—can be maintained. These ratios will be manually set depending on experience during the first few weeks of operation of the system, after which—unless the conditions change—they should need no further adjustment.

The air operated control instruments are said to have a straight line control characteristic regardless of the amount of reset.

In each water heater and heating coil assembly, the control governs the flow of water bypassed around the heater. Thus, as the outdoor temperature drops, the control functions through a diaphragm valve in the supply line to the heater to force more hot water



Typical heater setup, showing the 30 gpm circulator, bypass arrangement, heater itself, and the supply and return lines serving a floor heating coil in one of the bays. The placards (not legible in this reproduction) explain the function of each item of equipment, and were used during an inspection tour of the new plant last month

from the heater to the circuit. When the outdoor temperature goes up, the supply to the water heater is throttled down, forcing water through the bypass instead of the heater. The heater, in turn, is equipped with its own controls to modulate the gas flow to it depending upon the demand for hot water. The 30 gpm hot water circulators in each of the individual "unit" systems operate continuously.

According to Mr. Colby, the design and installation of a radiant heating system is very simple and "anybody can do it." All you need to know, he says, is the occupancy use of the space involved and that a sufficient amount of surface is installed. There are many factors of safety (such as raising the water temperature) which make absolutely accurate figuring unnecessary. Two of the bays in the plant area, and a pitted area where freight cars are loaded and unloaded inside the plant, have no heating coil installations, and it is felt that these areas will get enough heat from adjoining bays so that comfort will be maintained.

In general, the ground was brought to grade, the floor pipe coils put down, wire mesh laid on top, and the 6 in. concrete slab poured. In one section, strips of insulation were placed beneath the pipe coils, and thermocouples were installed in the ground below to determine heat flow downward from the floor coils with and without the insulating strips.

At each end of the plant is a large outdoor parking area, and the walkways to these areas have pipe coils embedded in the concrete slab for snow melting.

The only mechanical ventilation in the plant area is that due to process requirements, and the air to supply these needs is heated either by steam from the process boiler installation or by direct gas firing. Some consideration has been given to the use of a pond of water on the roof to reduce solar heat gain in the summertime.

Government Reports on Underground Plants

Construction and design features of 116 underground installations, built by the Germans to minimize the effect of Allied air bombing on industry, are described in a series of three reports on sale by the Office of Technical Services, Department of Commerce. One of the reports contains complete descriptions of Hitler's Berchtesgaden retreat.

The reports were prepared by the U. S. Army investigators in Europe.

Several outstanding structures built from the ground down were found. However, for the most part, the investigators report, the Germans simply remodeled existing facilities—mines, caves, railroad tunnels, and beer halls—to house underground factories. Nothing novel in this type of improvised construction was noted.

PB-25638 (photostat, $26; microfilm, $8; 384 pages) describes nine major underground factories in Germany and Austria.

The most unique constructions were four semi-underground Messerschmitt factories at Munich. The roof was a 280 ft span of reinforced concrete, 10 ft thick. The roof arch was poured on a 163,000 cu yd gravel fill which had to be excavated after the concrete had set. Five- and eight-story buildings, made of precast columns, beams, and floor slabs, were constructed underneath such roof arches. Heating and ventilating facilities were more elaborate than in any other underground facilities examined.

In Austria the investigators found another large Messerschmitt factory covering an underground floor area of 12 acres. The factory had been completed in 13 months and had produced about 1000 jet propelled planes by the end of the war. At Minden, Germany, an underground oil refinery had been completed but was not yet in production at the end of the war.

The report contains a comprehensive description of each of the nine installations, illustrated by photographs and drawings.

Report PB-25639 (photostat, $5; microfilm, $2; 70 pages) describes German techniques for remodeling existing underground establishments to accommodate factory installations in Germany. Although these underground installations would not meet American standards of production efficiency, the report states, they did produce a large quantity of war materiel.

Very little consideration was given to the comfort of personnel working in these plants. Fans and filters for removing dust were antiquated, sanitary facilities were inadequate, and heat was provided only to prevent air condensation on the machinery. The close quarters in which the work was carried on must have created a nerve shattering din, say the investigators. Installations found in salt, potassium, gypsum, and iron mines, in highway and railway tunnels, caves, brewery cellars, and vaults are described in the report. Drawings and diagrams are included.

Report PB-27779 (photostat, $15; microfilm, $5; 211 pages) describes installations of general interest, including power plants, shelters, munitions plants, oil depots, tunnels, communications centers, fortifications, and headquarters and residences. Complete descriptions of Hitler's Berchtesgaden retreat and Nazi Party Secretary Bormann's underground residence at Munich, with data on utilities, design, ventilation, and camouflage, are contained in this report. Drawings are included.

German attempts to move industry underground did not result in construction designs new to American engineers, according to the reports. However, because of shortages of material and use of second-rate equipment, together with the use of slave labor, some of the construction methods are unusual. In spite of handicaps, the Germans made rapid headway with the underground construction program, and had there been more advance planning for the structures the war might have been prolonged.

Several drawings mentioned but not contained in the reports because of their bulk, are on file in the Office of the Chief of Engineers, War Department, Washington, D. C.

Orders for the reports should be addressed to the Office of Technical Services, Department of Commerce, Washington 25, D. C., and should be accompanied by check or money order, payable to the Treasurer of the United States.

Fundamental Characteristics of the CENTRIFUGAL PUMP

By John J. Woolfenden
Smith, Hinchman & Grylls, Inc.
Architects and Engineers

[with sketches by the author]

SINCE the use of the centrifugal pump has steadily increased until it is commonly employed for various purposes formerly served by pumps of other types and since—although it is a relatively simple piece of apparatus—there are many who have had no particular opportunity or necessity of becoming familiar with its characteristics, a brief discussion of the more fundamental of them may be of interest.

It is necessary to distinguish the true centrifugal type from the more recently developed radial blade pump, which has quite different characteristics and which is commonly known as a turbine pump. There may be some confusion caused by this nomenclature as a centrifugal pump equipped with guide vanes to control the direction of flow of the water delivered by the impeller has often been called a turbine pump.

Two Essential Parts

The general arrangement and appearance of the centrifugal pump is so familiar that a detailed description of its construction is unnecessary. It will suffice to say that it consists of two essential parts, an impeller which by its rotation induces the requisite centrifugal force in the fluid handled and a stationary casing of suitable design enclosing the impeller and provided with inlet (or suction) and outlet (or discharge) openings for the ingress and the egress of the fluid handled. All other parts are auxiliary to one or both of these two.

Although the pump will handle liquids other than water at ordinary temperatures, the customary manufacturers' catalog ratings are based upon the pumping of clear, cold water and as this medium will serve sufficiently well for illustrative purposes, it will be used as a basis for the present discussion.

The water is admitted to the impeller at its central portion. The rotary motion of the impeller is imparted to it and the resulting centrifugal force causes it to move outward to the impeller periphery, whence it is discharged into the casing. Theoretically, the velocity of the water leaving the impeller is the same as the peripheral speed of the impeller. It is important to note that the water is discharged from the impeller uniformly around the entire periphery.

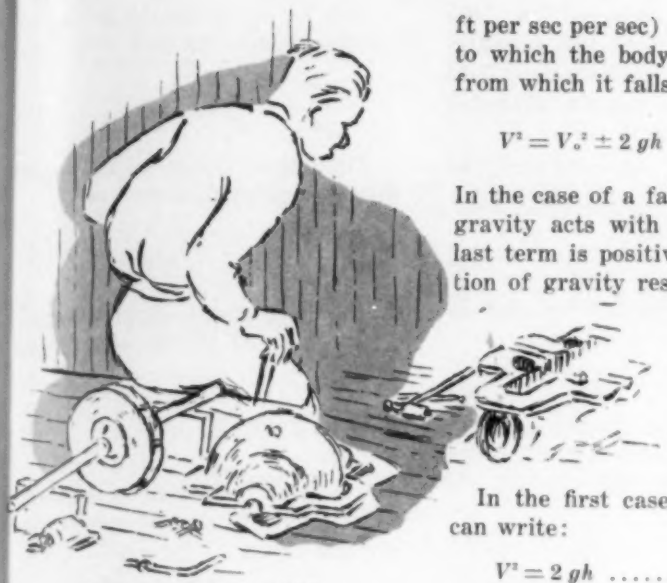
The inside of the casing, in its simplest type, forms a volute, which —beginning with zero cross sectional area at a point adjacent to the pump discharge—continuously increases in area in the direction of the impeller rotation until it reaches its maximum at the pump discharge opening, to which it delivers the water.

"Taking the air"

Owing to the fact that the direction of the flow in the volute at any point does not coincide with the direction of the water discharged by the impeller, which is the resultant due to the rotation and the centrifugal force, it is obvious that a properly designed system of guide vanes is necessary if this discrepancy is to be corrected. However, that is a refinement which has to do with efficiency of operation and does not affect the fundamental principles; moreover, numerous pumps have been designed that operate at satisfactorily high efficiencies in spite of the fact that guide vanes have not been provided. Furthermore, the angle of any fixed guide vanes would be correct for only one impeller speed.

Similar to Centrifugal Fan

From the foregoing description, it will be apparent that the basic principles involved are similar to those with which we have to deal in the case of centrifugal fans; in fact, they are identical. This is very apparent in the case of the centrifugal high pressure fan. Such fans have been designed for a discharge pressure of as much as 32 oz per sq in. and their internal construction is remarkably similar to that of a double suction, enclosed impeller, centrifugal pump.



"Where is its head?"

The essential differences between the design of a centrifugal fan and a centrifugal pump are due to the facts that the density of air is much less than that of water and that air is an elastic medium—whereas water, for all practical purposes, is not. Furthermore, the expansion of air per degree of temperature is much greater than that of water.

The fundamental equations, often called the "affinity equations," are the same for the fan as for the pump.

Peripheral Speed of Impeller

The peripheral speed of a pump impeller can be expressed, in feet per second, by the following equation, in which d is the diameter of the impeller in inches and n the rpm:

$$V = \frac{\pi d n}{12 \times 60} \quad \ldots\ldots\ldots\ldots [1]$$

According to the laws of falling bodies, neglecting the effect of air resistance, an object projected vertically upward with an initial velocity V_0 will under the influence of gravity decelerate until at the highest point of its path its final velocity V is zero. It will then start to fall, beginning with an initial velocity of $V_0 = 0$ and will return to its starting point with a final velocity V due to the acceleration of gravity which will be numerically equal to its original initial velocity. Hence we may say, letting g represent the acceleration due to gravity (32.16 ft per sec per sec) and h the height to which the body is projected or from which it falls:

$$V^2 = V_0^2 \pm 2gh \quad \ldots\ldots\ldots\ldots [2]$$

In the case of a falling body, when gravity acts with the motion, the last term is positive; when the action of gravity results in deceleration, as in the case of a body projected upward, the last term is negative.

In the first case, if $V_0 = 0$, we can write:

$$V^2 = 2gh \quad \ldots\ldots\ldots\ldots [3]$$

while in the second case, if $V = 0$, we have

$$V_0^2 = 2gh \quad \ldots\ldots\ldots\ldots [3a]$$

For the same values of h, these values of V and V_0 are obviously numerically equal; hence, for simplicity, it is customary to use Equation 3, although we are actually dealing with the case represented by Equation 3a.

Theoretical Maximum Head

Since, theoretically, the water leaves the impeller at the peripheral velocity, the value of h obtained from Equation 3 by substituting for V the peripheral velocity of the impeller in feet per second represents the maximum head in feet which that particular impeller, at the peripheral speed V, is theoretically capable of developing, or the maximum height to which (neglecting friction) it is theoretically capable of raising the water.

From this it is obvious that, if h is assumed, V is fixed—and that if either the impeller diameter or its rpm is selected, the other quantity follows. Thus, substituting in Equation 3 the value of V from Equation 1, we have:

$$\left(\frac{\pi d n}{12 \times 60}\right)^2 = 2gh \quad \ldots\ldots\ldots [4]$$

If we solve this equation for h we have:

$$h = \frac{(\pi d n)^2}{2g(12 \times 60)^2} \quad \ldots\ldots\ldots [5]$$

from which, by substituting the values of g and π and performing the indicated operations, we find that the denominator of this equation becomes very nearly 1840^2, and the equation may be written:

$$h = \left(\frac{d n}{1840}\right)^2 \quad \ldots\ldots\ldots\ldots [6]$$

in which d is in inches and n in rpm. It is important to remember that, in the case of direct drive by a-c motors, the actual and not the synchronous rpm should be used.

The value of h thus obtained is the maximum head the impeller is theoretically capable of developing. The maximum discharge head, however, is not necessarily the "shutoff" head, or head at which the pump ceases to discharge, since the design may be such that the internal losses at shutoff may be greater than those due to water flow under discharge conditions; hence, the actual maximum head measured at the pump discharge may correspond to a definite discharge capacity.

It is to be noted that neither Equation 6 nor any of the preceding ones is in any way related to the *capacity* of the pump, since no expression for mass is included; these equations are concerned with the peripheral velocity of the impeller, only.

As regards the relationship between the impeller design and the pump capacity, it may be said that for a given peripheral impeller speed the pump capacity will be a function of the impeller width (not necessarily in exact proportion, as other influences must be considered).

ENDEAVORING TO BE as clear and brief as possible, Mr. Woolfenden reviews here the fundamental characteristics of that "standard" item of piping, heating and air conditioning equipment—the centrifugal pump. His article is not intended for the pump designer, but simply to give the man who uses centrifugal pumps (and who may not be overly familiar with the theory involved) a little light on "the nature of the beast." He not only tells what its characteristics are, but explains why.

"Now, where does this go?"

The "affinity equations" were mentioned above; these are expressions representing certain basic characteristics referred to the peripheral speed of the impeller. They are given as Equations 7-9; in each case, the impeller diameter remains unchanged:

The *capacity varies directly as the speed, or

$$\frac{gpm_1}{gpm_2} = \frac{rpm_1}{rpm_2} \quad \text{.....[7]}$$

The †head (pressure) varies as the square of the speed, or

$$\frac{h_1}{h_2} = \frac{rpm_1^2}{rpm_2^2} \quad \text{.....[8]}$$

The horsepower required varies as the cube of the speed, or

$$\frac{hp_1}{hp_2} = \frac{rpm_1^3}{rpm_2^3} \quad \text{.....[9]}$$

That Equation 9 must result from the relationships expressed by Equations 7 and 8 is obvious, since the weight of the liquid handled is proportional to the capacity.

As a corollary to these, we can say that, the speed remaining constant:

The *capacity varies as the impeller diameter, or

$$\frac{gpm_1}{gpm_2} = \frac{d_1}{d_2} \quad \text{.....[10]}$$

The †head varies as the square of the impeller diameter, or

$$\frac{h_1}{h_2} = \frac{d_1^2}{d_2^2} \quad \text{.....[11]}$$

The horsepower required varies as the cube of the impeller diameter, or

$$\frac{hp_1}{hp_2} = \frac{d_1^3}{d_2^3} \quad \text{.....[12]}$$

The results obtained by the use of Equations 7-12 are, of course, theoretical values.

*The capacity which the pump will deliver against the head corresponding to the speed upon which the new capacity is based.

†The head corresponding to the new speed upon which the new capacity is based. If the actual head is not that corresponding to the new speed, the capacity will not be that corresponding to the new speed. It is important to note that, unless otherwise stated, it is assumed in the subsequent discussion that, when a change of speed is stipulated, the new head will be that corresponding to the new speed.

Proposes Utility Heat Pump Program

At a meeting held in Birmingham last month, further progress was made in the program proposed by Philip Sporn to further the development of the "heat pump" as applied to all-electric, year 'round heating and cooling and domestic water heating. At this meeting, a group of some 60 representatives of electric light and power companies in 17 southern and southwestern states, extending from Virginia to Arizona, were present to hear and discuss Mr. Sporn's proposals. This meeting, sponsored by the Southeastern Electric Exchange, was the second that Mr. Sporn has conducted.

The first meeting was attended by representatives of electric companies from the northeastern, middle Atlantic and north central states and was held in New York in the offices of the American Gas and Electric Service Corp., of which Mr. Sporn is executive vice president.

In papers read before meetings of the Southeastern Electric Exchange and the Association of Edison Illuminating Companies in April of this year, Mr. Sporn proposed that the business managed electric utilities throughout the United States join in a cooperative effort to encourage manufacturers to engineer and develop packaged "heat pump" units that "could be manufactured on a production basis and sold at realistic prices" by placing substantial orders. The conferences at New York and Birmingham were held to discuss means to accomplish this. Mr. Sporn has developed tentative specifications and hopes to be able to secure firm commitments for the purchase of 1000 air conditioning and heating units and 10,000 water heating units.

The specifications are expected to be used as a guide for the manufacture of heat pumps in quantity. The principle of the heat pump long has been known. The most familiar application to date of this principle has been in the field of refrigeration. When the equipment that is used to refrigerate is reversed to take heat from the outside and pump it into a space to be heated, it becomes a heat pump. So far, there has been installed a limited number of commercial applications of the heat pump using standard refrigeration equipment.

The specifications being developed contemplate an automatic, electric motor driven unit for homes and commercial establishments which will serve the double purpose of cooling and heating.

It is also believed that the application of the principles of the heat pump to a unit designed for water heating will result in an even lower cost of electric water heating than at present.

Mr. Sporn has found that a number of leading manufacturers are interested in the production of the equipment and, at the meetings already held, many of the electric companies attending have expressed a willingness to join in the program and to purchase and install experimental units when they have been produced.

Condensing Pressures for Air Conditioning

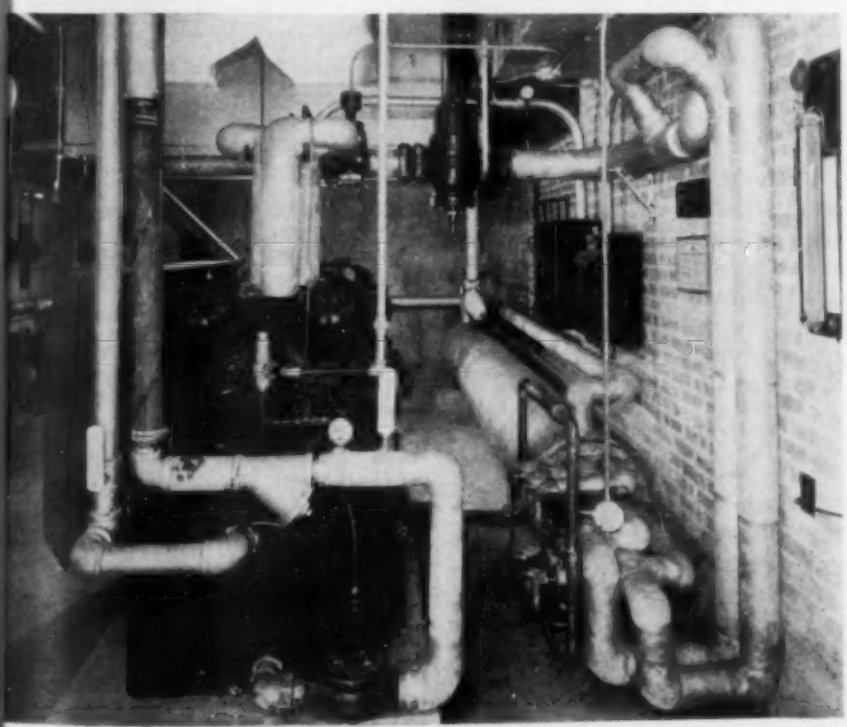
Andrew P. Boehmer Develops Method of Analyzing Operating Costs of "Freon-12" Condensing Units

INSTALLATION of a refrigeration condensing unit involves the important problem of economical operation. This is particularly true in air conditioning, where refrigeration equipment may be sizable and operating costs of considerable importance.

Among the items that loom large in the eyes of the operator are the cost of electrical energy used by the compressor motor and the cost of supplying the condensing water. These are inter-related in that—for normal operating conditions—an increase in water flow through the condenser reduces the condensing pressure, and the electric load on the compressor motor; conversely, a reduction in water flow results in increased motor load.

This paper develops an analytical approach to the operating cost problem for the purpose of determining the conditions of operation that will result in the minimum overall cost of both electrical energy used by the compressor motor, and the cost of the condensing water. Data on condensing units with reciprocating compressors ranging in size from 10 to 60 hp have been taken from published manufacturers' catalogs.

An increase in water flow through the condenser reduces the condensing pressure and the electric load on the compressor motor. The text explains how to determine the conditions of operation that will result in the minimum overall cost of both electric energy and condensing water



The total operating cost (C_t) is shown to be primarily the cost of electrical energy to the compressor motor (C_e), and the cost of the condensing water (C_w). Thus, $C_t = C_e + C_w$.

Cost of Electrical Energy to Compressor Motor

The cost of the electrical energy to operate the compressor per unit of refrigerating effect is dependent upon the condensing pressure (P_c), the evaporating or suction pressure (P_s), and the electric rate (E) when compressor efficiency, motor efficiency, and superheat of the suction vapor are held constant.

The ratio of condensing pressure to suction pressure, in lb per sq inch absolute, is important and is defined as

$$r = \frac{P_c}{P_s} \quad \text{.....[1]}$$

Theoretically, the energy required per ton of refrigeration (K_t) equals

$$K_t = \frac{12{,}000\ (h_4 - h_3)}{3413\ (h_2 - h_1)}$$

$$= \frac{3.52\ (h_4 - h_3)}{(h_2 - h_1)} \text{ kw per ton}$$

where h_1 = enthalpy of the refrigerant liquid entering the expansion valve; h_2 = enthalpy of the suction vapor at the compressor; and h_4 = enthalpy at condensing pressure and at same entropy as State 3.

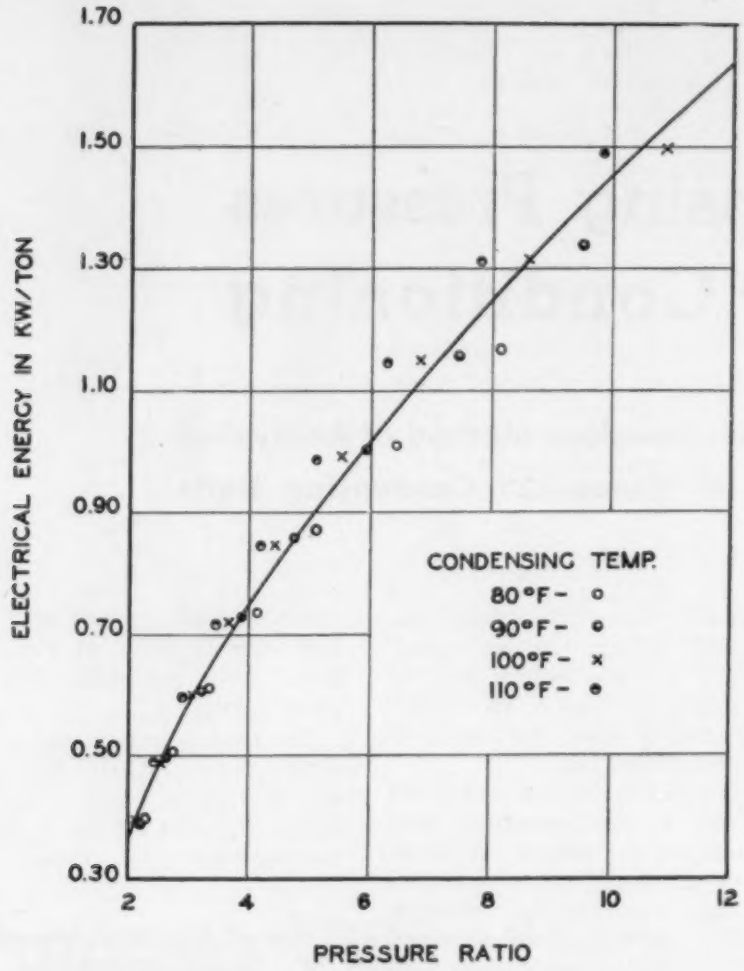


Fig. 1—Electrical energy required per ton of refrigeration for ideal cycle

Fig. 1 shows the variation of electrical energy required per ton of refrigeration with various pressure ratios using "Freon-12" as the refrigerant for the theoretically ideal cycle. Suction vapor was assumed to enter the compressor at a 10 F superheat above saturation at the suction pressure.

A sample computation for a point on this curve follows:

Assume $t_s = 40$ F $P_s = 51.7$ psia
$t_c = 90$ F $P_c = 114.3$ psia

Then

$$r = \frac{114.3}{51.7} = 2.25$$

$$K_i = \frac{3.52(90.45 - 84.19)}{(84.19 - 28.70)}$$

$$= 0.397 \text{ kw per ton}$$

The electrical energy required at a pressure ratio is affected by the evaporating pressure and the condensing pressure used to arrive at that ratio. This is indicated in Fig. 1 by marking each condensing temperature used in determining any value of pressure ratio. An average curve drawn through these points has the following equation:

$$K_i = 0.665\ r^{0.68} - 0.56 \text{ kw per ton}$$

Values of electrical energy for a compressor motor determined from this expression obviously will be low for actual applications because the data used in establishing the curve were based on a theoretically ideal cycle.

Study of manufacturers' catalogs reveals that the actual electric energy required (within the ranges herein considered) is approximately 1.79 times that required theoretically. The actual electrical energy required (K_a) for a compressor is thus:

$$K_a = 1.79\ (0.665r^{0.68} - 0.56)$$

$$K_a = 1.19r^{0.68} - 1.00 \text{ kw per ton}$$

The electric cost for operation of the compressor is the energy requirement times the electric rate.

$$C_e = E\ (K_a) = E(1.19r^{0.68} - 1.00)$$
cents per hr per ton . . . [2]

where E = electric rate in cents per kwhr, including demand charge.

Cost of Condensing Water

The quantity of condensing water required depends upon the sum of the refrigerating effect and the heat equivalent of the work of compressing the refrigerant vapor, neglecting heat transfer from the compressor and the discharge line to the condenser.

For 1 ton of refrigeration, the heat removed by the condensing water equals

$$Q_c = 12{,}000 + 3413e_m\ (1.19r^{0.68} - 1.00)$$
Btu per hr per ton

where e_m = motor efficiency as a decimal.

The actual electrical energy required to operate a compressor is greater than that theoretically required by virtue of losses not only in the compressor but in the electric motor as well. The condenser, however, does not remove the heat equivalent of the loss of electrical energy in the motor; thus the motor efficiency must be eliminated from the computation of the heat removed by the condenser water. (This does not apply to hermetic units or other encased equipment.) Efficiencies of motors have been assumed as approximately 85 per cent. The heat removed in the condenser thus becomes

$$Q_c = 12{,}000 + 3413\ (0.85)(1.19r^{0.68} - 1.00)$$

$$Q_c = 9080 + 3450r^{0.68} \text{ Btu per hr per ton}$$

The condensing water required in cubic feet per hour per ton of refrigeration is

$$M = \frac{9080 + 3450r^{0.68}}{62.4\ (t_2 - t_1)} \text{ cu ft per hr per ton}$$

where t_1 = water temperature entering the condenser, F; and t_2 = water temperature leaving the condenser, F.

The refrigerant condensing vapor temperature (t_c) has been assumed to be 5 F higher than the temperature of the water discharging from the condenser. While this is an arbitrarily chosen value, it checks reasonably well with manufacturers' published data. By substituting for t_2, the condensing water required may be expressed

$$M = \frac{9080 + 3450r^{0.68}}{62.4\ (t_c - 5 - t_1)} \text{ cu ft per hr per ton}$$

To make possible the mathematical differentiation of the above

IT IS POSSIBLE to determine the operating conditions of refrigeration condensing units of from 10 to 60 hp to give maximum operating economy by selecting condensing conditions of the unit which will result in the minimum total cost of electricity for the compressor motor and water for condensing. The condensing pressure for air conditioning installations may be established when the electric rate, water cost, and the temperature of the water supply to the condenser are known. The condensing water discharge temperature is the best criterion of economical operation, as this temperature is relatively independent of the operation of the condenser. Maximum economy of operation is maintained with varying condenser water supply temperatures if the quantity of water is varied to maintain a constant temperature difference through the condenser. . . . The author is assistant professor of M.E. at the University of Rochester.

expression it is necessary to establish t_c as a function of the pressure ratio. This is accomplished by first establishing the relation of t_c with P_c.

From Fig. 2, which is plotted from standard "Freon-12" tables, one may determine an empirical relation which exists between the condensing temperatures and condensing pressures within the limits shown. This may be written

$$t_c = 116\ (P_c)^{0.238} - 233$$

By substituting from Equation 1

$$P_c = rP_s$$

and introducing the water charge (W), the cost of water per ton of refrigeration equals

$$C_w = \frac{W(9080 + 3450r^{0.48})}{62.4[116(rP_s)^{0.238} - 238 - t_1]}$$

$$C_w = \frac{W(146 + 55.6r^{0.48})}{116(rP_s)^{0.238} - (238 + t_1)}$$

cents per ton per hr. . . . [3]

where W = water cost in cents per cubic foot.

The equation for the total operating cost, including the cost of electrical energy and the cost of the condensing water, may now be written as a function of the pressure ratio and a number of independent variables.

$$C_t = E(1.19r^{0.48} - 1.00) + \frac{W(146 + 55.6r^{0.48})}{116(rP_s)^{0.238} - (t_1 + 238)} \quad \ldots\ldots\ldots\ldots\ldots [4]$$

This equation may be used to estimate the cost of operation of any "Freon-12" unit within the scope of this paper if all of the variables have been established. Computations of actual costs are simplified by substituting t_c for its equivalent.

$$C_t = E(1.19r^{0.48} - 1.00) + \frac{W(146 + 55.6r^{0.48})}{t_c - 5 - t_1} \quad \ldots\ldots [5]$$

Computations of total costs of operation throughout the following analysis are made with the aid of Equation 5.

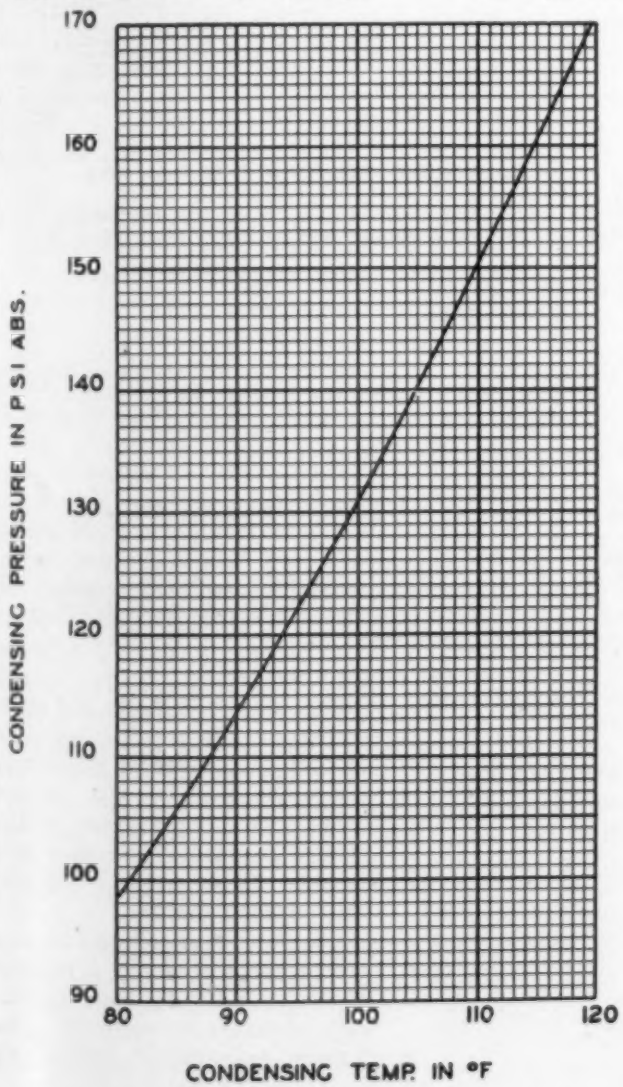


Fig. 2 — Pressure-temperature relationship of "Freon-12"

Economic Condensing Pressure

Equation 4 was established in terms of the pressure ratio to make it possible to take the partial derivative of the total cost with respect to that variable. By setting the first derivative equal to zero, a value of pressure ratio may be determined for which the total cost is a minimum for any specific conditions of water cost, electric rate, suction pressure, and condensing inlet water temperature.

Setting $(dC_t/dr) = 0$ and letting $L = (E/W)$ the expression becomes

$$0.571L = \frac{\dfrac{3650(P_s)^{0.48}}{(rP_s)^{0.284}} - 1680(rP_s)^{0.238} + 26.3(t_1 + 238)}{[116(rP_s)^{0.238} - (238 + t_1)]^2}$$

By substituting P_c for rP_s the final form of the equation becomes

$$L = \frac{\dfrac{(6380\ P_s^{0.48})}{(P_c)^{0.284}} - 2940\ (P_c)^{0.238} + 46\ (t_1 + 238)}{[116(P_c)^{0.238} - (t_1 + 238)]^2} \quad \ldots [6]$$

By a substitution of values for the pressure ratio in terms of the condensing pressure, it becomes

possible to compute the minimum value of L for any conditions of P_c, P_s, and t_1. Fig. 3 was plotted as a solution of this equation. By making use of this family of curves, the selection of an economical condensing pressure for any particular value of L, at the evaporating temperature shown, is a simple matter.

The data in Fig. 3 for establishing condensing pressures were based on discharge water temperatures assuming commercial scale factors when selecting condenser surface—that is, by assuming heat transfer based on slightly scaled tubes, and with sufficient condenser surface to reduce condensing temperatures to within 5 F of the temperature of the water leaving the condenser. It will be shown later that even though the condition of the condenser is not known, the most economical condition of operation may be determined by use of the discharge water temperature rather than the condensing pressure.

The latter, however, may be considered as a criterion for most installations of standard manufacturers' equipment in the sizes considered.

Example of Use of Economic Condensing Pressures

To demonstrate use of Fig. 3 and to show the effect of variation of cost due to improper selection of condensing pressure, a sample problem is considered.

Assume that a 10 ton "Freon" refrigeration unit operating at air conditioning back pressure is in a locality where the electric cost is 1.5 cents per kwhr and the water cost is 50 cents per 1000 cu ft. Water inlet to the condensing unit is assumed to be 75 F. Under these conditions,

$$\frac{50}{1000} = 0.05 \text{ cents per cu ft}$$

$$L = \frac{1.5}{0.05} = 30$$

From Fig. 3, the correct operating head pressure is found to be 139 psia, or 124 psig. The pressure P_s at air conditioning coil evaporating temperatures is approximately 51.7 psia. Then,

$$r = \frac{P_c}{P_s} = \frac{139}{51.7} = 2.69$$

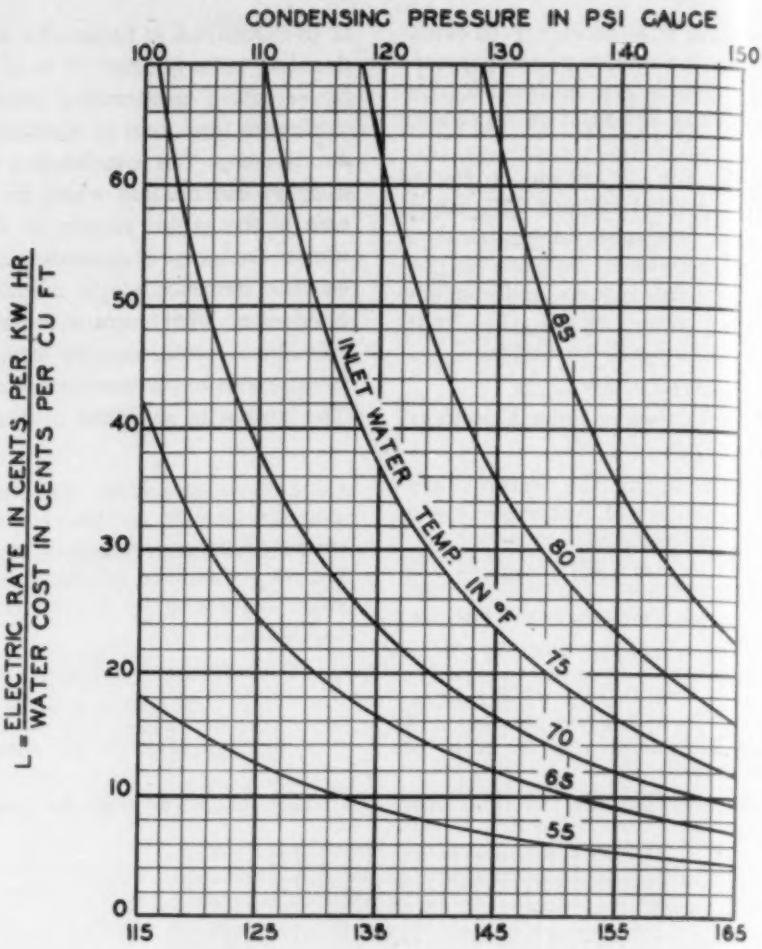


Fig. 3—Economic condensing pressures for "Freon-12" condensing units operating at air conditioning evaporating temperatures

The actual minimum cost of operation may be determined by using Equation 5:

$$C_t = 1.5\,[1.19\,(2.69)^{0.48} - 1] + \frac{0.05\,[146 + 55.6\,(2.69)^{0.48}]}{(104 - 5 - 75)}$$

$C_t = 1.37 + 0.49 = 1.86$ cents per hr per ton

This gives an operating cost of 18.6 cents per hr for 10 tons of refrigeration under the conditions assumed. The division of cost shows 13.7 cents for electrical energy and 4.9 cents for condensing water.

Selection of Other Than Most Economical Condensing Pressure

To determine the effect on total cost if a lower than economic head pressure is maintained, an assumption will be made of a head pressure of 132 psi instead of 139. The pressure ratio under these conditions is

$$r = \frac{132}{51.7} = 2.55$$

and the cost of operation becomes

$C_t = 1.30 + 0.58 = 1.88$ cents per hr per ton

Comparison of operating costs will be on a per ton basis as this will more readily cover the general case, and will make comparisons easier with actual published data for units of various sizes. The cost per ton of operating with a 132 psia head pressure is 1.3 cents for electricity, and 0.58 cents for water. Increasing the quantity of water through the condenser causes a decrease in cost of the electrical energy, but the reduced cost of electricity is more than offset by an increase in the cost of supplying the condensing water.

An increase in condensing pressure brought about by a reduction in condensing water supplied below that necessary to maintain a condensing pressure of 139 psi also

shows an increase in the cost of operation. To illustrate this, conditions will be assumed in which condensing water is reduced until the head pressure on the unit is 151 psia. Under these conditions,

$$r = \frac{151}{51.7} = 2.92$$

$C_t = 1.49 + 0.40 = 1.89$ cents per hr per ton

Use of insufficient water shows an increase in total cost above the minimum, due to the greater increase in electrical energy cost than the reduction in condensing water cost in operating under these conditions.

In an investigation of the variation in cost over a large range of condensing pressures with 75 F inlet water to the condenser, Fig. 4 was plotted. This figure shows that the cost of electrical energy increases as a straight line function of the head pressure while the condensing water cost decreases as the water is reduced. The total cost, being a sum of the two, starts at a high value, flattens out at the minimum, and reverses itself after passing the most economical head pressure.

Inasmuch as the total cost curve becomes almost horizontal at the minimum cost of operation, the selection of the economic head pressure is not particularly critical, although a major variation from the most economical pressure will show an appreciable increase in cost.

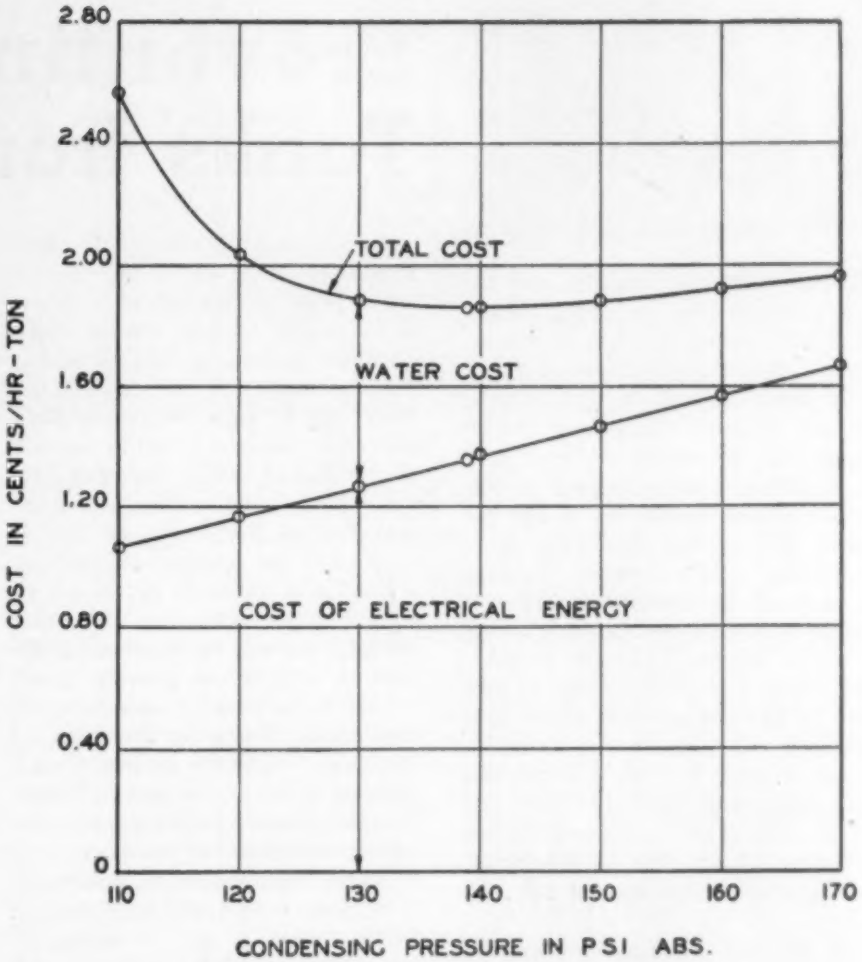


Fig. 4—Theoretical cost of operating condensing unit at various condensing pressures

Fig. 3 may be used in selecting operating conditions for units operating at evaporating temperatures above approximately 20 F. Lower evaporating temperatures usually require lower values of condensing pressures than those selected with the aid of Fig. 3.

[A second article by Professor Boehmer will analyze manufacturers' catalog data for economic condensing conditions, and will discuss selection based on temperature difference through the condenser, variation in operating cost with varying condenser operation, and effect on economic head pressures of varying inlet water temperature to condenser.]

No Stockpiling of Building Materials

No stockpiling of building materials in anticipation of future construction is permitted by the Civilian Production Administration.

Stockpiling of certain materials and equipment has been permitted under priorities regulation 32 (the inventory control regulation) to encourage an orderly transition from war production to the production of civilian goods. In view of the tremendous demand for building materials in the present intensive drive for new housing, CPA wants to make it clear that this provision of the regulation does not and was never intended to permit the stockpiling of building materials for construction purposes.

Two changes in the regulation were also announced. Due to expiration of the requisitioning act, which provided for the requisition of materials for defense purposes, reference to it has been deleted and the regulation now states that disposal of unneeded materials acquired or made for use and not for sale or resale is subject to the special sales provisions of priorities regulation 13. Special sales are sales of materials or components acquired or manufactured for use and not for resale in their present form.

Because of the current shortage of electrical wiring devices needed in the housing program, these items have been added to Table 1 of the regulation, thereby imposing a 45 day inventory limitation or a practicable minimum working inventory, whichever is less.

SAM LEWIS' PAGE

Preventing Buried Tanks from Floating

ENGINEERS occasionally are required to compute the added weight that will be necessary to prevent a buried tank from floating up out of its "grave" in case the tank is empty and in case the ground water level rises around it.

If the tank is submerged in the water soaked earth, one might assume that it has displaced water equivalent to its volume. Imagine a tank of approximately 8000 gal capacity. Such a tank could be about 7 ft in diameter and 28 ft long. The metal would weigh around 12,000 lb. The surface area of the cylinder, ignoring the heads, would be about 616 sq ft. The area at the meridian would be 7 × 28, or 196 sq ft. One might assume that the effective lifting area of a horizontal cylinder is the area at the meridian.

If the tank contains 8000 gallons of water, it would weigh about 8000 × 8.3 or 66,400 lb and if full of fuel oil or water, the tank assuredly would stay in place even if entirely submerged below ground water level.

If the tank is buried, with 3 ft of earth above its top, if it is empty, and if there is such a flood that ground water stands a foot deep above the surface of the ground, the conundrum is to compute how much added weight shall be attached to the tank to prevent its rising.

At first thought, one engineer suggested that the displaced water outside the tank all would be putting its shoulder under the tank to heave it loose—in other words, 66,400 lb less the weight of the tank, 12,000 lb or 54,400 lb effective weight when submerged, would have to be attached to the tank to hold it down.

Another engineer suggested that the pressure of the water outside the tank has a bearing on the situation. The water under the bottom of the tank is under a head of 11 ft.

A column of water 11 ft high weighs 62.35 × 11 = 685 lb and could push up against the bottom of the tank (if the bottom were flat) at the rate of 685 lb per sq ft. The water at the top of the tank is 4 ft high, and could push downward against a similar surface at the rate of 249 lb per sq ft. The water at the sides of the tank at its meridian is 3+1+3½ or 7½ ft deep and is pushing inward at about 7.5 × 62.35 or 467 lb per sq ft, but is not doing any particular work in the way of shouldering the tank up out of the ground.

The water pushing upward at 45 deg points, below the meridian of the tank, evidently is partly neutralized by the water pushing downward at similar angles against the upper hemisphere of the tank.

Another consideration, not exactly subject to definite computation, is that when a tank is buried in the ground, even though the ground may be saturated with water, the tank is not going to be shouldered out without lifting along with itself a very much greater volume of earth than that which lies directly above the area described by the length and width of the tank.

In other words, it is likely that the earth would have to be upheaved over a distance about double the width of the tank. Again, if the earth is saturated with water, the effective holding-down influence of any added weight may be expected to be the difference between the weight in air and the weight when under water. Thus a cubic foot of concrete in air may weigh 130 lb, but under water will weigh only about 68 lb.

We have had many arguments and considerable amusement over this little problem. It seems to be one that may interest some of our mathematically minded friends.

Our tentative solution is that the empty tank weighs in water 10,500 lb. The water displaced by the tank weighs 67,000 lb. The difference is 56,500 lb. The concrete pad weighs under water, 68 lb per cu ft. 56,500/68 = 830 cu ft. Then 830/28 indicates a pad of concrete 28 ft long, 30 ft wide and 1 ft thick, or 28 ft long, 15 ft wide and 2 ft thick for a tank 7 ft in diameter and 28 ft long. This bulk of concrete seems rather startling.

There is little doubt in my mind but that the empty tank with this much concrete fastened to it would sink to the bottom if dropped into Lake Michigan.

The speculations also lead one to give some serious consideration to placing large oil tanks entirely above ground, even though they may have to be surrounded with fireproof enclosures and dikes to limit the oil flow in case they should leak.

IN THIS regular feature, informal comment on heating, piping, and air conditioning matters is given by Samuel R. Lewis, consulting mechanical engineer, a member of HPAC's board of consulting and contributing editors. This month he gives some practical pointers on computing the added weight required to prevent a buried tank—such as a fuel oil tank—from floating when it is empty.

Methods of Making

PIPING FLEXIBILITY

Analyses

GRAPHICAL METHOD

By M. J. Fish, Advisory Engineer, Combustion Engineering Co., Inc.

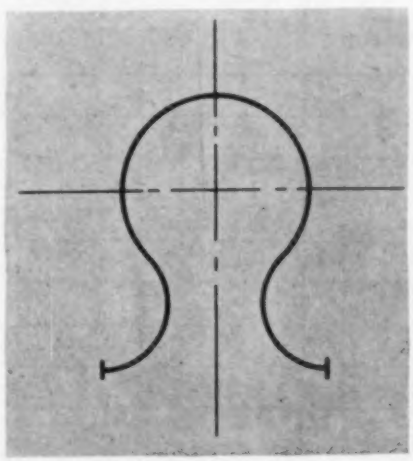
In 1910, Dr. A. Bantlin, of Stuttgart, published the results of tests on short radius, double offset pipe bends similar to that of Fig. 1. These experiments were made on seamless steel tubing and on cast iron bends. The results with seamless steel tubing showed deflections as much as five times those which could be accounted for by the ordinary formula for bending of curved solid bars. The tests on the cast iron bends did not show such discrepancies. Bantlin ascribed the discrepancy between the observed and the calculated displacements in the steel tubes to wrinkles and corrugations in the pipe wall during the process of bending.

Shortly thereafter, Dr. Th. v. Kárman developed formulas by the method of minimum energy which explained the discrepancy by the flattening of the circular pipe section, which takes an oval form when loaded. This flattening is due to bending while the pipe is subjected to expansion forces and not to the process of manufacture. Von Kárman's equation was found independently by Hovgaard, who verified it by tests made on full size specimens at the Massachusetts Institute of Technology. These two investigators accounted for the flattening effect at the arcs by the introduction of a "Flexibility Factor."

Tangent Lines for Arcs

In the method described in this article, tangent lines are substituted for the arcs. This gives sufficient accuracy in view of the various uncertainties present in the problem. Since the introduction of the flexibility factor at the arcs lowers the factor of safety of the pipe line and since the influence of the arcs is small, if the ratio of the straight lengths of pipe to the arcs is large, the factor has been neglected in the following work. The pipe line will be considered to lie wholly in one plane with its ends fully restrained and subjected to a uniform temperature change. As such, it constitutes a statically indeterminate system. When a pipe line is restrained from moving, the stresses and end reactions are increased. Therefore, the assumption of full restraint is on the safe side.

Fig. 1—Tests on short radius, double offset bends made in 1910 showed deflections as much as five times those which could be accounted for by the ordinary formula for bending of curved solid bars. The discrepancy was explained by the flattening of the circular pipe section, which takes an oval form when loaded



To review some elementary facts before analyzing the actual problem, consider a beam restrained at the ends and uniformly loaded, as in Fig. 2a. The elastic line or curve of the neutral surface after bending will have the two points of inflection at a distance of $0.211L$ from the ends. The bending moment diagram of the beam, Fig. 2b, shows that the value of the bending moment at the points of inflection is zero. It should be noticed that due to the restraint at the ends, the total change in the angle of flexure (the angle θ made by a tangent to the elastic curve with the horizontal) over the length of the beam is zero.

Fundamental Beam Theory Equation

The fundamental equation of the beam theory is:

$$\frac{d\theta}{dl} = \frac{M}{EI} \quad \text{.....[1]}$$

where dl = an elementary length of the neutral surface of the beam; $d\theta$ = the change in slope of the tangent to the neutral surface in the length dl; M = the beam's bending moment, which is usually a function of x only; E = the modulus of elasticity of the material; and I = the moment of inertia of the cross section of the beam.

Integrating Equation 1:

$$\theta_L - \theta_0 = \int_0^L \frac{M}{EI}\, dl \quad \text{.....[2]}$$

where θ_L = the angle of flexure at the right end of the beam; and θ_0 = the angle of flexure at the left end of the beam.

The value of I will be taken as constant but E varies with the temperature according to the formula which is given for mild steel in Fig. 3. However, once the temperature t in degrees Fahrenheit is given, E is constant and will be taken so in the analysis. Equation 2 can be written as:

$$\theta_L - \theta_0 = \frac{1}{EI}\int_0^L Mdl \quad \ldots\ldots [3]$$

It is seen that the expression

$$\int_0^L Mdl$$

is the area of the moment diagram and it follows that the difference in slope between two points on a beam of uniform section is the area of the moment diagram between these points divided by EI. Therefore, it follows that the moment area is a direct measure of the angle of flexure along the beam.

Assumptions in Analyzing Piping

In the analysis of a pipe line, the following assumptions are made: The cross section of the tubes is circular and uniform; local stresses due to fabrication have been relieved; the dead weight of the members is neglected; the influence of the longitudinal forces (direct tension or compression) is neglected; the energy due to direct shear is negligible; the bending moment at any point of the members is not affected by the deflection of the members; and the material obeys Hooke's law.

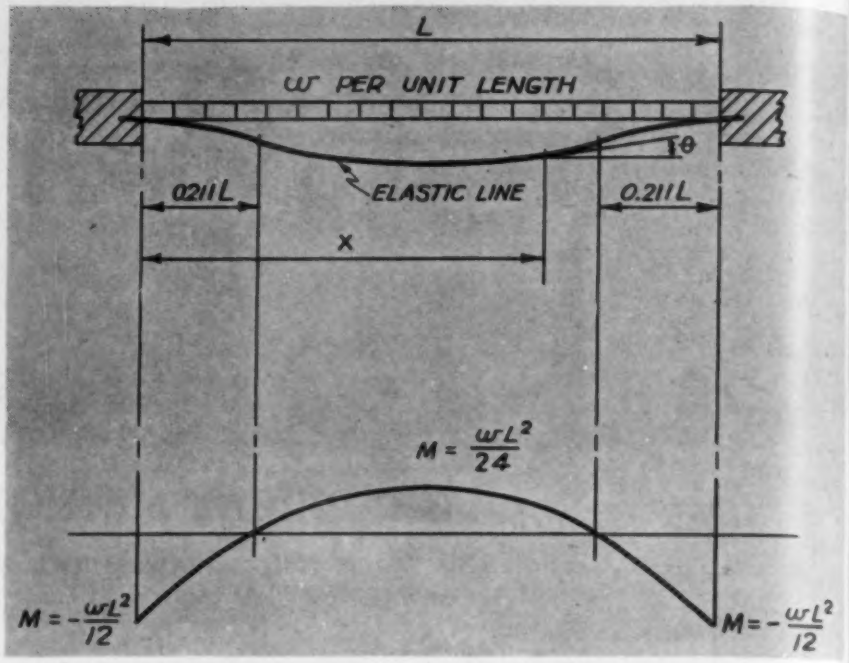
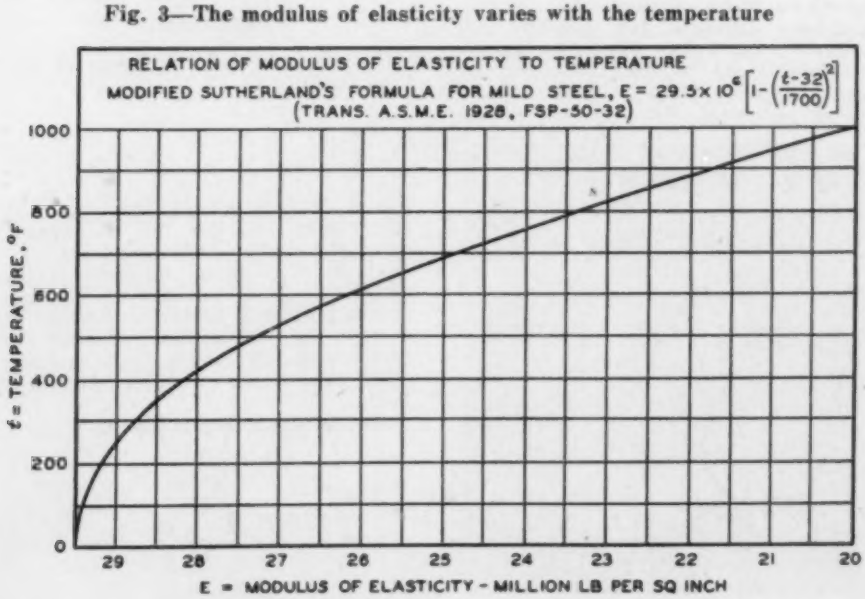


Fig. 2a (above)—Beam restrained at the ends and uniformly loaded . . . Fig. 2b (below)—Bending moment diagram for the beam of Fig. 2a

Now, consider the symmetrical pipe line in Fig. 4, consisting of the two equal vertical legs, L_2, and the horizontal leg, L_1. The arcs have been replaced by ideal corners in order to simplify the demonstration. Let the ends A and B of the system move along parallel restrained paths under the action of the forces P. This means that the ends A and B will undergo the motion of translation but are restrained from rotation. The members of the system will flex as shown by the dotted line in Fig. 4. Since there is no rotation of the ends, the sum of all the angles of flexure (positive and negative) must equal zero. Therefore, it follows from Equation 3 that the sum of the moment areas is also zero. This latter condition is true only when the moment area diagrams are taken about an axis parallel to the direction of translation of the restrained ends and through the centroid of the system. The moment at any arbitrary point of a member is equal to the force P times the lever arm h, where the latter is the perpendicular distance from the above mentioned axis through the centroid to the arbitrary point. Therefore, it follows that the moment is zero where the gravity axis intercepts the members of the system and since it is well known that where the moment is zero there is a point of inflection of the elastic line, then the points of inflection are determined by the intersection of the members with the gravity axis.

The flanges in the pipe line should be placed as near as possible to the points of inflection in order to prevent excessive bending on the bolts with consequent leaking at the gaskets.

Fig. 3—The modulus of elasticity varies with the temperature



Proof of Foregoing Statements

To prove and illustrate the foregoing statements, draw, in Fig. 4, a line parallel to AB and at a distance x from member L_1. Taking moments about this line, the moment in L_1 equals Px and is denoted

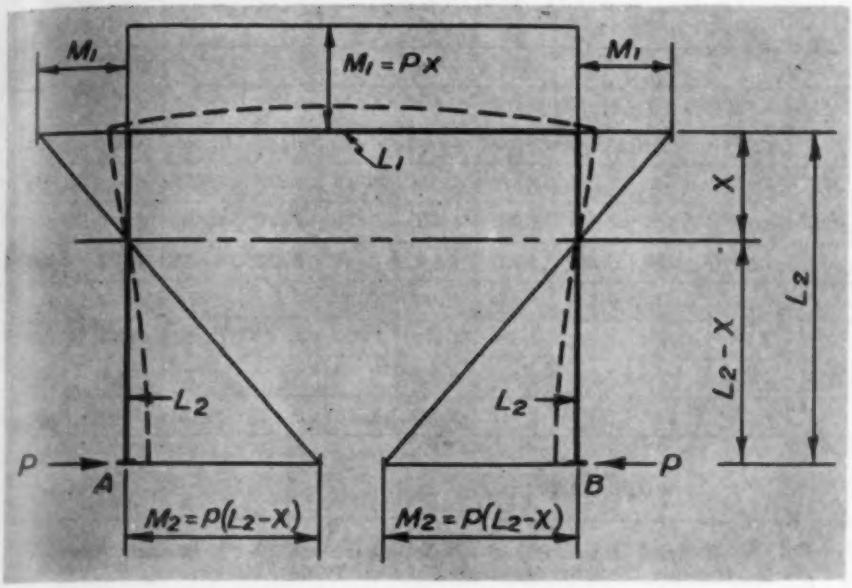


Fig. 4—Symmetrical pipe line consisting of two equal vertical legs and a horizontal leg

by M_1; the moment in x equals Px; and the moment in $L_2 - x$ equals $P(L_2 - x)$ and is denoted by M_2.

Since the total change in the angle of flexure in the members equals the moment area divided by EI, then:

$$\Delta\theta_{L_1} = \frac{PxL_1}{EI}$$

$$2 \cdot \Delta\theta_x = 2 \cdot \frac{Px^2}{2EI} = \frac{Px^2}{EI}$$

$$2 \cdot \Delta\theta_{(L_2-x)} = -2\left[\frac{P(L_2-x)^2}{2EI}\right] = \frac{P(L_2-x)^2}{EI}$$

where $\Delta\theta$, with the subscript, is the change in the angle of flexure of the member denoted by the subscript letter, and

$$\Delta\theta_{L_1} + 2\Delta\theta_x - 2\Delta\theta_{(L_2-x)} = 0 \quad \ldots [4]$$

Substituting the values above, Equation 4 now reads:

$$\frac{PxL_1}{EI} + \frac{Px^2}{EI} - \frac{P(L_2-x)^2}{EI} = 0 \ldots [5]$$

Expanding Equation 5:

$$\frac{P}{EI}[xL_1 + x^2 - L_2^2 + 2L_2x - x^2] = 0$$

Therefore

$$x = \frac{L_2^2}{L_1 + 2L_2} \quad \ldots [6]$$

The right hand side of Equation 6 is the expression of the center of gravity of the pipe line. This proves the statement that for zero moment area, the moment area diagrams must be taken about an axis parallel to the direction of translation and through the center of gravity. If the arcs are replaced by tangents instead of ideal corners, the effect is to change the position of the center of gravity of the system and the numerical values of the bending moments but the method of procedure will be the same.

Having determined the points of inflection of the system, the internal potential energy of bending is to be equated to the external work producing the bending. From the theory of elasticity, it is well known that the internal work of bending is:

$$W = \frac{1}{2EI}\int_0^L M^2 dl \quad \ldots [7]$$

Equation 7 may be written:

$$W = \frac{P^2}{2EI}\int_0^L h^2 dl \quad \ldots [8]$$

where h is the variable lever arm of the bending moment and dl is an elementary length of the member.

Referring to Fig. 6, in the integration of Equation 8:

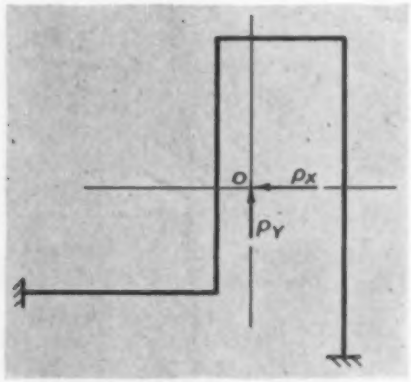


Fig. 5—With an asymmetric piping system, the solution consists of the evaluation of the component forces P_x and P_y

$$W = \frac{P^2}{2EI}\int_0^L \left(\frac{H_a l}{L}\right)^2 dl = \frac{P^2 H_a^2 L}{6EI},$$

for a triangle.

$$W = \frac{P^2 H_b^2 L}{2EI}, \text{ for a rectangle.}$$

$$W = \frac{P^2 L}{6EI}(H_c^2 + H_c H_d + H_d^2), \text{ for a trapezoid.} \ldots [9]$$

In Fig. 4, denote the value of x as found by Equation 6 by H_1 and $L_2 - x$ by H_2. Then, from Equations 8 and 9:

$$W = \frac{P^2}{2EI}\left[H_1^2 L_1 + 2\frac{H_1^2 H_1}{3} + 2\frac{H_2^2 H_2}{3}\right]$$

$$= \frac{P\Delta}{2} \quad \ldots [10]$$

where P is the external force and Δ, the change in distance between A and B due to force P.

Therefore:

$$P = \frac{\Delta EI}{\left[H_1^2 L_1 + \frac{2}{3}H_1^3 + \frac{2}{3}H_2^3\right]} \quad \ldots [11]$$

If the method of least work or minimum energy is applied to Equation 10, then from:

THERE ARE numerous methods of analyzing the flexibility of piping systems and a great amount of material on this important subject has been published. In the June HPAC, Sabin Crocker and Arthur McCutchan reviewed the various methods, and suggested that the advocates of the different schemes each briefly outline the merits of his method and show the solution of two problems; in this way, the reader can judge for himself the suitability of the different calculation procedures in his own piping work. . . . S. W. Spielvogel followed this suggestion in July, explaining the essentials of the elastic center method and solving the two problems presented by Messrs. Crocker and McCutchan by means of it. This month, Mr. Fish writes on the graphical method. Others will follow.

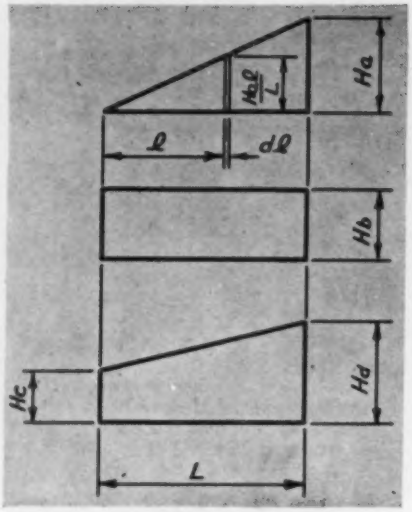


Fig. 6—This diagram is used in developing Equations 9 of the text

	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°
0°	—	—	—	—	—	—	—	0.00	0.067	0.154
100°	0.240	0.318	0.397	0.474	0.551	0.631	0.710	0.790	0.870	0.951
200°	1.033	1.113	1.194	1.278	1.362	1.444	1.525	1.608	1.692	1.777
300°	1.861	1.947	2.032	2.118	2.204	2.288	2.371	2.462	2.553	2.635
400°	2.717	2.813	2.908	2.995	3.082	3.176	3.270	3.356	3.442	3.540
500°	3.638	3.734	3.829	3.921	4.012	4.107	4.202	4.298	4.393	4.491
600°	4.589	4.684	4.779	4.874	4.969	5.071	5.173	5.268	5.362	5.466
700°	5.570	5.664	5.767	5.872	5.977	6.076	6.175	6.282	6.388	6.490
800°	6.592	6.694	6.806	6.905	7.004	7.117	7.230	7.335	7.440	7.548
900°	7.655	7.771	7.887	7.992	8.097	8.207	8.317	8.428	8.538	8.651

$$L_t = L_o\left[1 + a\left(\frac{t-32}{1000}\right) + b\left(\frac{t-32}{1000}\right)^2\right]$$

L_t = Length @ Temp. t, F (in feet)
L_o = Length @ 32 F (in feet)
$a = 0.006212$ $b = 0.001623$

Fig. 7—Thermal expansion of steel pipe (from 70 F) in inches per 100 ft (formula from Smithsonian Physical Tables, 7th rev. ed., p. 218)

Fig. 8—Z bend with welding elbows, used as illustrative example

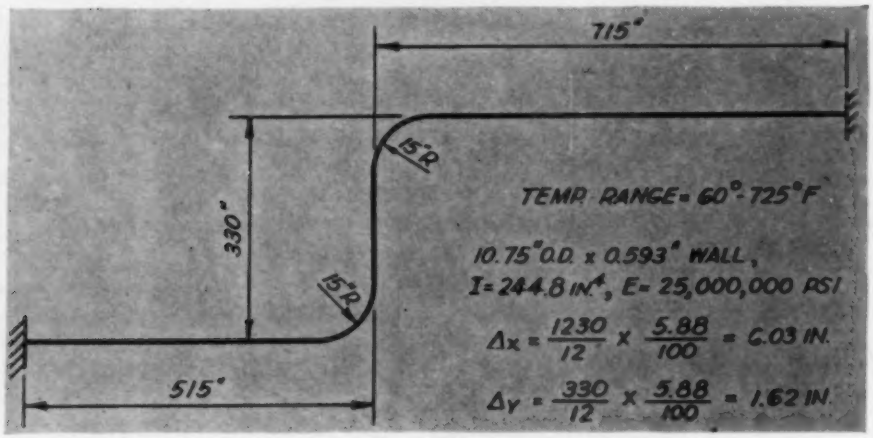
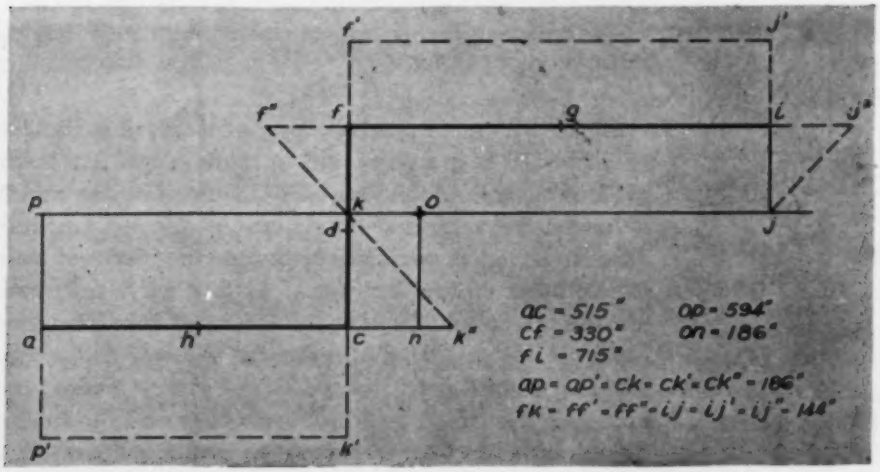


Fig. 9—Diagram used for solving example illustrated in Fig. 8



$$W = \frac{P^2}{2EI}\left[L_1^2x^2 + \frac{2}{3}x^3 + \frac{2}{3}(L_2 - x)^3\right]$$

it follows that:

$$\frac{dW}{dx} = \frac{P^2}{2EI}[2L_1^2x + 2x^2 - 2(L_2 - x)^2] = 0$$

and:

$$x = \frac{L_2^2}{L_1 + 2L_2}$$

which checks the value in Equation 6.

In the more general case of the asymmetric system, the solution consists of the evaluation of the component forces P_x and P_y at the centroid O. (Fig. 5.) The simultaneous equations are:

$$AP_x - CP_y = EI\Delta x$$
$$-CP_x + BP_y = EI\Delta y \quad \ldots.[12]$$

where A is the denominator of the right side of Equation 11 with the axis in the X-direction through the centroid; B is the denominator of the right side of Equation 11 with the axis in the Y-direction through the centroid; and C is the sum of the respective products of the component lengths of the pipe multiplied by the horizontal and vertical distances of their centers from the centroid, with the proper algebraic signs.

Illustrative Example

Consider the pipe line of Fig. 8 which is fixed at the ends, with the dimensions and data as given. Since the arcs are small in comparison to the straight lengths of the pipe, tangents are not used and the original pipe line is replaced by *acfi* of Fig. 9. Locate the centers of *ac*, *cf*, and *fi* at *h*, *d*, and *g*, respective-

Fig. 10—Diagram used for solving example illustrated in Fig. 8

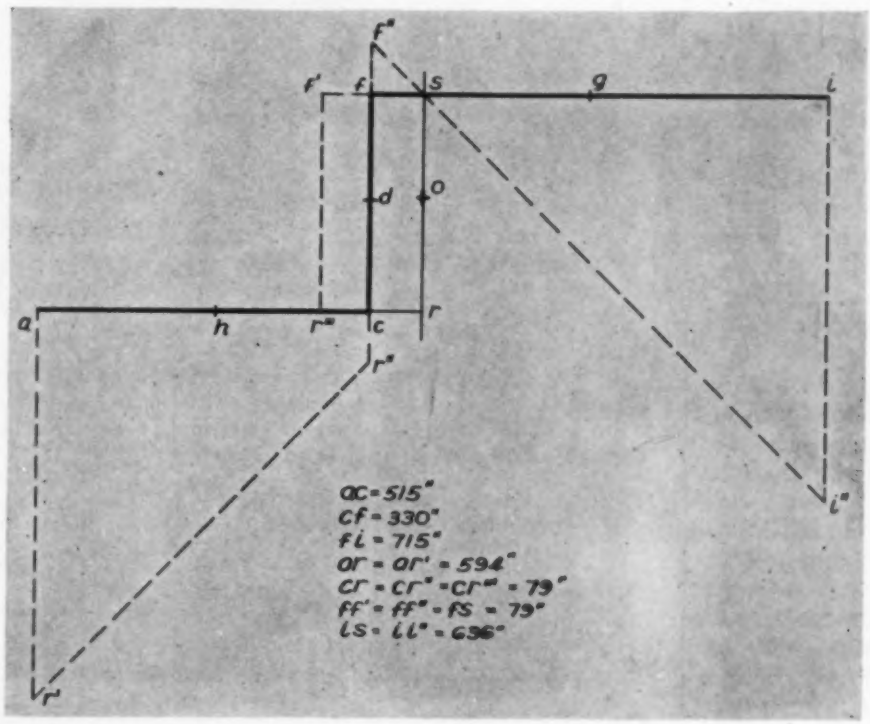
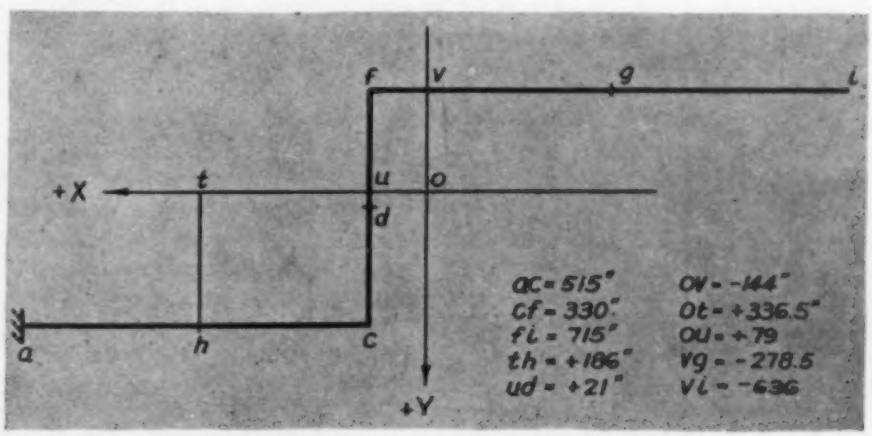


Fig. 11—Diagram used for solving example illustrated in Fig. 8



ly. Then find the center of gravity of the pipe line by taking moments about *ac* and a line through *a* which is perpendicular to *ac*, as follows:

Moments about ac

ac × 0 =	515 × 0 =	0
cf × 165 =	330 × 165 =	54,450
fi × 330 =	715 × 330 =	235,950
	1560	290,400

Moments about perpendicular line through a

ac × 257.5 =	515 × 257.5 =	132,613
cf × 515 =	330 × 515 =	169,950
fi × 872.5 =	715 × 872.5 =	623,838
	1560	926,401

$\frac{290,400}{1560} = 186$ in. $\frac{926,401}{1560} = 594$ in.

This locates point *O*, the centroid, such that *on* = 186 in. and *op* = 594 in.

In order to determine coefficient *A*, draw the horizontal axis through *O* (Fig. 9). The moment area diagram consists of the two rectangles *ap'k'c*, *ff'j'i*, and the three triangles *ck"k*, *ff"k*, *oj"j*. Therefore, the value of the denominator of the right side of Equation 11 can be found as follows:

Rectangle $ap'k'c = (186)^2 \times 515 = 17,816,940$

Triangle $ck"k = \frac{(186)^2}{3} \times 186 = 2,144,951$

Triangle $ff"k = \frac{(144)^2}{3} \times 144 = 995,327$

Rectangle $ff'j'i = (144)^2 \times 715 = 14,826,240$

Triangle $ij"j = \frac{(144)^2}{3} \times 144 = 995,327$

$A = 36,778,785$

To determine the coefficient *B*, draw the vertical axis through *O* (Fig. 10). The moment area diagram consists of the trapezoid *ar'r"c*, the rectangle *ff'r'"c*, and the two triangles *ff"s*, *ii"s*. Therefore, the value of the denominator of the right side of equation 11 can be found as follows:

Trapezoid $ar'r"c = \frac{[(594)^2 + 594 \times 79 + (79)^2]}{3} \times 515 = 69,697,180$

Rectangle $ff'r'"c = (79)^2 \times 330 = 2,059,530$

Triangle $ff"s = \frac{(79)^2}{3} \times 79 = 164,346$

Triangle $ii"s = \frac{(636)^2}{3} \times 636 = 85,753,152$

$B = 157,674,208$

In Fig. 11, if the point *a* is considered fixed and the end *i* allowed to expand freely, it must be displaced to the left and down in order to bring it back to its original position. Assume these directions to be positive, Fig. 11. The coefficient *C* is found below:

Member *ac* = 515 × 186 × 336.5	= 32,233,335
Member *cf* = 330 × 79 × 21	= 547,470
Member *fi* = 715 × (−278.5) × (−144)	= 28,674,360
	$C = 61,455,165$

The simultaneous Equations 12

$$36,778,785\ P_x - 61,455,165\ P_y = 36,903,600,000$$
$$-61,455,165\ P_x + 157,674,208\ P_y = 9,914,400,000$$

have as their solution $P_x = 3178$ lb and $P_y = 1301$ lb.

In actual practice, the pipe line does not have to be redrawn when finding the coefficients *A*, *B*, and *C*. It has been done in the above demonstration for the sake of clarity.

The bending moments are:

3178 × 144 − 1301 × 636 =	−369,804 lb-in. at *i*
3178 × 144 + 1301 × 79 =	560,411 lb-in. at *f*
−3178 × 186 + 1301 × 79 =	−488,329 lb-in. at *c*
−3178 × 186 + 1301 × 594 =	181,686 lb-in. at *a*

The maximum fiber stress at point *f* is:

$$\frac{560,411 \times (10.75/2)}{244.8} = 12,300 \text{ psi.}$$

COMPRESSOR CEILINGS INCREASED

Increases in reseller's ceilings for compressors and condensing units 5 hp or less in capacity and repair and service parts therefor, warm air furnaces, floor and wall furnaces, cast iron radiation and accessories, specified tanks and vessels, brass plumbing fixture supply fittings and trimmings and waste fittings and trimmings, were announced August 23 by the Office of Price Administration to comply with the new price control law's requirement on distributor margins.

As a result, it is estimated that consumers will pay from 1 to 3 per cent more for compressor and condensing units and warm air furnaces; 1.5 per cent more for the brass plumbing fixture supply fittings and tanks and vessels and about 1 per cent more for cast iron radiation.

The series of orders is similar to several others already issued by OPA on other building materials where resellers had been given dollar-and-cent increases on increased costs resulting to them from increases granted their suppliers after April 1.

"FOREIGN AFFAIRS"

Cyril Tasker Reviews Proposed New Heating and Ventilating Standards

New British Report Suggests Background, Top-Up Heating

ANY readers of this magazine who spent one or more recent winters in the British Isles will—on the slightest provocation—give their unrestrained opinion of heating there. After 16 years on the American continent, the author needs little persuasion to agree.

Improved standards of warmth for homes in Britain recommended in the recent report of the heating and ventilation (reconstruction) committee of the Building Research Board for the Ministry of Works are still below those used in most parts of the American continent. They are, however, a distinct improvement over conditions found in most pre-war British homes.

The report is one of a series of Postwar Building Studies. The objectives of the committee (which included several prominent members of the Institution of Heating and Ventilating Engineers among its members) were to review existing scientific information and practice in England and abroad, and to make recommendations for postwar building practice and future research. A subcommittee, appointed to investigate the desirability of developing district heating schemes in Britain, has not yet reported.

Though the factors in heating that intimately affect the householder were given full consideration, certain broader considerations involving the wisest use of the national coal resources had to be taken into account. The report, available through the British Information Services, Rockefeller Center, New York City, is well worth studying by those interested in future developments in heating. The urgent need for improvement in Great Britain, together with the importance of using coal resources wisely, may provide a stimulus for the development of efficient equipment for low cost housing which will merit our keeping closely in touch with progress abroad.

"Brightly Burning Fire"

In Great Britain, as in America, the yearly cost of fuel for heating, for cooking, and for domestic hot water has become of increasing interest. Increased fuel costs have turned attention to its more efficient use. This in turn has awakened interest in better construction and efficient and economical methods of heating.

Early in 1942, as a preliminary to the main study, the Ministry of Works, in cooperation with the gas and electrical industries, made an inquiry into the heating of dwellings occupied by lower income groups. The results of this inquiry, supported by numerous tables of statistics, form part of the report here under review. Though this is not the place to discuss the results, it may be stated that they had a distinct influence on the findings of the main committee.

For example, the survey showed that the brightly burning fire in the livingroom fireplace is still the focal point for the ordinary social activities of the average English family; yet it showed that many housewives would like some scheme of central heating even if it did not include an open fire. The analysis of the replies to inquiries about central heating showed that it was liked less in the colder regions of the country than in the warmer regions. The personal experience of the author and his family bears out the fact that the psychological comfort derived from a fire is more appreciated in the colder and less sunny parts of the country. The inquiry also showed that there was a big demand for the warming of upstairs rooms and for easy means of heating water.

Statistics on the annual domestic fuel consumption per person and per house in Great Britain, the United States, and Germany (all for prewar years) showed some interesting if surprising figures, which are given in Table 1.

Persons with experience of both America and Great Britain will probably suggest at once that if the Btu consumption per person in the U.S.A. for domestic purposes is less than 8 per cent greater than it is in Great Britain, the efficiency of application in America must greatly exceed that in Great Britain.

Table 1—Annual domestic fuel consumption in Great Britain, the United States, and Germany (1936 and 1938)
(All figures are millions)

	Britain	U.S.A.	Germany
Number of households.	12½	32	18
Population	46	129	66
Btu per house	149	171	88
Btu per person	40	43	24

ALTHOUGH PRIMARILY devoted to the heating and ventilation of dwellings, the recent report of the heating and ventilation committee of the Building Research Board for the Ministry of Works considers comfort standards in general and discusses methods of more or less interest in connection with all types of buildings. Mr. Tasker is director of research of the American Society of Heating and Ventilating Engineers, and is well-posted on current British practice.

Basic Requirements for Heating and Ventilation

In framing their recommendations the committee states that the comfort and health of the majority of persons were the major considerations and that neither comfort nor health should be overemphasized at the expense of the other. "Temperature in the home should be controllable and people should be instructed how to use and *regulate* the appliances provided." (The italics are the author's.) In line with the housing commission of the League of Nations health organization, which expressed the opinion that "harm may be done to the faculty of adaptation of the human body by the tendency . . . for a continuous increase in and a too steady maintenance of the (full comfort) temperature of dwellings," the committee felt that it was possible "to have too much comfort."

The report recognizes the importance of air movement, of temperature gradient, of radiation, and of stuffiness in considering desirable warmth conditions in living rooms. Speaking of radiation, it is suggested that ceiling heating should not be used as the sole source of heat in rooms much less than 12 ft high.

Stuffiness may be experienced when the air temperature is much greater than the mean radiant temperature; conversely, when the air temperature is less than the mean radiant temperature a feeling of freshness is noticed. This may be due to the greater effect of variations of air movement on the convection heat loss from the body when the air temperature is low.

Discussing heating by radiation plus convection, the committee suggests that comfortable conditions would be obtained with an air temperature of 63 to 67 F and a mean radiant (wall) temperature about 3 deg higher (66 to 70 F.) These conditions give equivalent temperatures of 62 to 66 deg. (An *equivalent* temperature of 70 deg corresponds approximately to an *effective* temperature of 66 deg.)

When high temperature radiant heating is being used (that is, open coal or coke fires and gas or electric fires), the radiation into the room is not uniform. To obtain a comfortably warm room with small high temperature sources of radiation, the report recommends air temperatures of 58 to 62 F, wall temperatures the same, and additional (undirectional) radiation up to about 40 Btu per sq ft per hr on the surface of the body. The *equivalent* temperatures under such conditions will be from 62 to 66 deg.

Background Heating

Discussing heating by convection alone, the committee suggests that it is preferable, and gives a feeling of greater freshness to most people, to use convection heating to bring the air and wall temperatures to say 55 to 60 F (that is, *background heating*, in which the whole room is warmed to a temperature somewhat lower than the comfortable level) and to use a radiant source to "top-up" to the equivalent temperatures of 62 to 66 deg required for adequate warmth.

This concept of background heating and the use of sources of radiant heat to "top-up" is receiving much attention in Great Britain and is certainly worthy of some attention for low cost housing in America. The first reaction may be that "it wouldn't work" in America, but are we so sure? There are many differences in the construction of British and American low cost housing, but an impartial study might reveal possibilities of improving conditions in that 50 per cent of homes now heated by stoves.

Warmth Conditions in Bedrooms

The British climate and the porous wall construction of British houses prevalent before the war

Table 2—**Relative heat requirements for different occupancy periods**

Daily period of occupation hours	Total heat input required (as percentage of that for continuous heating over 24 hr)
2	25-35
5	35-45
10	60-75
15	75-90

made it more than likely that unheated rooms, and clothing and furniture stored in them, would be damp in the winter. Many British bedrooms are unheated (as many GI's remember only too well). The committee recommends facilities for obtaining an equivalent temperature of 50 to 55 deg during dressing and undressing and for preventing the room temperature dropping below 50 F during the night. To take care of illness there should be means of warming at least one bedroom to an equivalent temperature of 65 deg at low cost and with a minimum of labor and inconvenience.

An equivalent temperature of 60 deg is considered adequate for kitchens and temperatures of 50 to 55 deg for halls, passages, and bathrooms.

Ventilation Requirements

Early in the report the committee recommends that the living-room, the kitchen, and possibly one bedroom should be provided with a flue suitable for solid fuel appliances and, therefore, of not less than 50 sq in. cross sectional area, and that other bedrooms be provided with flues having a minimum cross sectional area of 30 sq in. Though such flues would not be required by central heating or electric radiators, the committee feels that they will (provided they are kept open) insure the removal of vitiated air from the rooms.

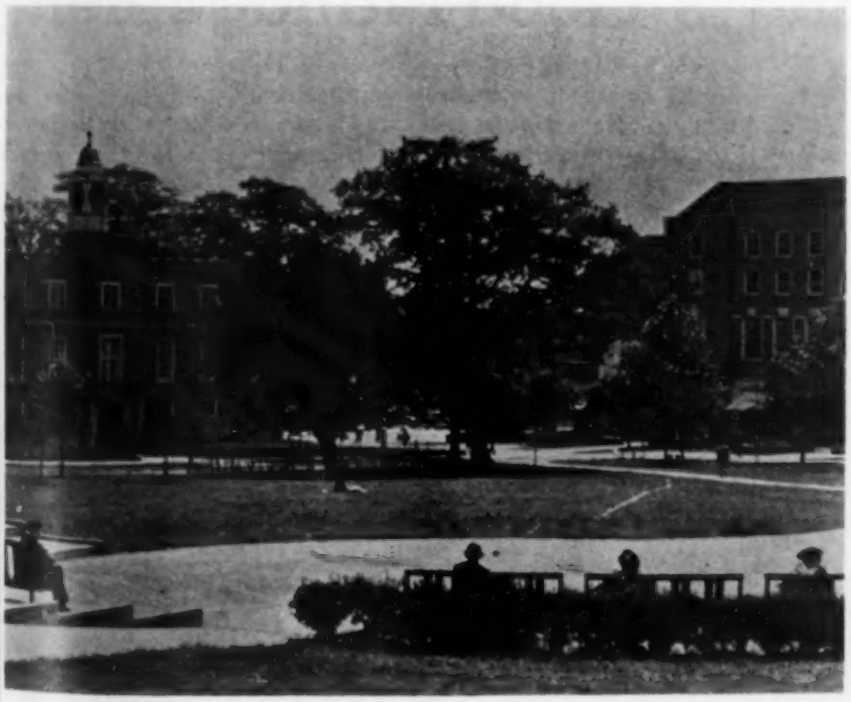
After studying all available evidence on the minimum amount of ventilation required for health, the committee adopted a figure of 600 cfh (10 cfm) per person as the normal minimum requirement, and suggested that to avoid excessive heating costs it be not greatly exceeded in winter.

In a room with a normal chimney with any method of heating in operation, and provided the flue is not restricted, sufficient ventilation will usually be obtained to satisfy this requirement.

Factors Affecting Amount of Heat Needed

Climate and Degree Days

The degree day used by British heating engineers is calculated to a base temperature of 60 F. Most of the southern part of England has an annual degree day total on this basis of 3500 - 4000 while in the north of England and most of Scotland the total varies from 4500 to 5500.



Urgent need of heating improvements in Great Britain, and the importance of using British coal resources wisely, merits our keeping in close touch with progress there

For the purpose of considering background heating to 50 F the report uses a base temperature of 45 F and under such conditions the London district has only 600 degree days per year. At a base temperature of 60 F, however, (and a corresponding room temperature of 65 F) the same area has 3750 degree days per year. Corresponding figures for the north of England are about 750 and 4750.

From weather data available in this country the degree days for various standards of background heating could readily be calculated. Calculations of this kind might well be a preliminary to any study of the possibility of using such schemes in America.

Intermittent Heating

In discussing intermittent heating, the committee states that the conditions of warmth assumed in the report may be reached in about 1½ to 2 hr in an ordinary room warmed daily from cold by radiant sources, the hourly heat input during this period being two to three times the hourly input required for maintenance of the steady state. These figures can only be approximate for there are so many and varied factors that enter into intermittent heating.

The committee gave a table (reproduced here as Table 2) of approximate relative heat requirements for different periods of occupancy expressed as percentages of the requirement for continuous heating over the 24 hr period.

In rooms lined with wood paneling, plywood, fiberboards, etc., a comfortable condition is obtained much sooner than in a room with a normal plaster finish.

Heat Transmission Coefficients

Many pre-War I British houses had exterior walls consisting of 9 in. of brickwork plastered one side, for which the thermal transmission (*U* factor) was 0.43 Btu per hr per sq ft per F. Since 1918 there has been an increasingly large use of 11 in. cavity brickwork walls plastered on one side. In many cases

the air space in such a wall has openings to the inside of the room and to the outdoors, thereby permitting ventilation of both room and cavity. This type of construction has a *U* factor of 0.34, similar to an uninsulated brick veneer frame wall with ½ in. gypsum sheathing and ⅜ in. gypsum lath, plastered, as used in American construction (ASHVE Guide, 1946, p. 127, wall No. 28). When the 11 in. cavity wall is unventilated, the *U* factor is 0.30. Many of the low cost houses erected in Britain are in semi-detached pairs, with the living rooms adjacent. When the party wall is in cavity construction, there is fairly good sound insulation between the living rooms.

Partition walls of 4½ in. brick, plastered both sides, used in England have about the same *U* factor as a hollow concrete block wall, plastered both sides, as used in this country. Tile roofs are extensively used in Great Britain, and a tile roof on battens with a plaster ceiling will have a *U* factor of 0.56. If boarded and felted and a ½ in. fiberboard ceiling is used, this is reduced to 0.22.

Ventilation

Wall ventilators were once used to a great extent in England. They are no longer considered a suitable or satisfactory means to provide the minimum ventilation requirements for a room. Some idea of the ventilation (or infiltration) effect produced by different methods of heating on the heat requirements of a warmed living room are given in Table 3.

Table 3—Effect of the ventilation produced by different methods of heating on the heat requirements of a warmed living room

Type of heating, with open fireplace and normal flue	Rate of air change		Heat required, Btu per hr per F difference between inside and outside, to warm ventilating air
	cfh	Air changes per hour	
Coal fire	7,000	4.5	133
Gas fire	4,800	3.1	91
Electric fire standing on hearth under flue	4,200	2.7	80
Hot water radiators	3,100	2.0	59
Restricted ventilation at rate of 10 cfm per person (4 persons)	2,400	1.6	46

Examples are worked out in the report to show the heat requirements with "background" heating of 50 F for 24 hr and "topping up" of various rooms to 55 F, 60 F, and 65 F for various parts of the day, for a terrace house of typical plan in the south of England (3750 degree days on the 60 F degree day base). The yearly heat losses in millions of Btu are given as 24.8, 18.85, and 16.26 with poor insulation, with a medium standard of insulation, and with a recommended standard of insulation, respectively. The significance of the part played by ventilation may be seen from the fact that even restricted ventilation (10 cfm per person) accounts for about 17 per cent, 23 per cent, and 26 per cent of the heat loss respectively. Any greater ventilation rate (for example, due to open coal fires) would greatly increase both the total heat loss and that proportion due to infiltration.

Insulation

Discussing insulation, the committee suggests that as an aid to economical heating and more efficient use of the nation's fuel resources external walls should be insulated to give a *U* factor no greater than 0.20 for any part of the house and no greater than 0.15 for the walls of the living room; that ground floors should not have *U* factors greater than 0.15, and roofs and top floor ceilings should have factors not above 0.20.

Air Conditioning Is 'Good Management'

AIR CONDITIONING and soundproofing are important factors in a busy airline reservation office, in the opinion of United Air Lines officials —and psychologically are rated as top morale builders.

That is why, in localities where air conditioning is an "accepted need," United is equipping its major reservations offices—often called the "pulse" of the airlines—with air circulating and cooling units.

The phrase, "accepted need," is explained by J. D. Crichton, United's superintendent of plant engineering, as embracing localities where summer temperatures are normally hot and humid. In such areas, both cooling and circulating units are installed. At other points, only air circulating units are used.

The company has made complete air conditioning installations in its reservations offices at New York City, Washington, Cleveland, Chicago, Omaha, and Los Angeles, and plans to so equip new offices in Detroit and Boston. In other localities —such as Oakland, San Francisco, Seattle, Portland, and Denver—air circulating units are made a part of new, modern reservations offices.

As explained by Mr. Crichton, United is endeavoring to provide the best working conditions for its reservations personnel.

"We all recognize that the job of answering a busy reservations telephone is confining, to say the least," says Mr. Crichton. "To help keep our reservations agents' dispositions pleasant, it is certainly good management to make their surroundings as pleasant as possible.

"With air conditioning we have cleaner offices through elimination of outside dirt and noise. That latter item is really important because there's no use in soundproofing an office if the window is to be opened so that streetcar and automobile noises can interfere with telephone conversations."

In the larger offices, such as at Chicago, 15 ton units are installed, but in such places as telephone switchboard rooms and smaller offices, United uses package units of 3 and 5 hp to eliminate duct work. United's engineers explain that the density of occupancy in a normal reservations office makes necessary above normal capacities.

Good Air Distribution Is Vital in Airplane Heating, Ventilating

Maurice Martin, Senior Research Engineer, Republic Aviation Corp., Discusses Air Supply and Distribution

PROPER AIR distribution is of vital importance in any air conditioning or ventilating system, but is especially so in the case of the airplane. Due to the large temperature drop between outside air and cabin, the inside wall temperature is very much below the average temperature in the cabin. Only a small percentage of the total sensible body heat is removed by convection. About 60 per cent is removed by radiation from the body to the surrounding walls, etc.

Radiation Loss and Its Effect on Comfort

The effective temperature—i.e., hypothetical temperature used as an indication of a relative feeling of comfort—may be lowered 5 deg or more by the cold walls (1)*. The concept of comfort being associated with an air temperature of 70 F breaks down entirely when the heated space is one, as an airplane cabin, where the air and surface temperatures may have widely different values (2).

The fundamental radiation equation is:

$$q_r = \sum_{n=1}^{n=n} (F_{os}\, s\, A_s) \left[\frac{1}{A_o} \int_{A_s} \int_{A_o} \frac{\cos \phi_s \cos \phi_o \, dA_s dA_o}{\pi r_s^2} \right] \left[\left(\frac{T_o}{100} \right)^4 - \left(\frac{T_s}{100} \right)^4 \right]$$

q_r = net energy loss, radiation from an occupant to the enclosing surfaces, Btu per hr, when enclosure consists of n different surfaces, each of area A_s and absolute temperature T_s;
A_o = effective area of occupant, surface at absolute temperature T_o;
s = Stefan Boltzmann constant = 0.172;
r_s = distance from center of area A_s to midpoint of occupant's projected area;
ϕ_s = angle between r_s and normal to A_s;
ϕ_o = angle between r_s and normal to A_o;
F_{os} = emissivity factor of A_s and A_o.

*Numbers in parentheses refer to the bibliography at the end of this article.

The first factor in brackets is the shape factor of the occupant F_{os}, and using a radiant transfer factor:

$$h_r = s \left[\left(\frac{T_o}{100} \right)^4 - \left(\frac{T_s}{100} \right)^4 \right] \Big/ (t_o - t_s) \cong 1.03$$

results in a simplified form:

$$q_r = \sum_{n=1}^{n=n} 1.03\, F_{os} A_s F_{os}\, (t_o - t_s)$$

t_o = average temperature of exposed surface of body.

Using known values of radiant heat loss and convective heat loss from the body, it is found that a 1 F temperature change in the surround is equivalent to an opposite air temperature change of 1.5 F. This calculation is based on an assumption of a homogeneous surround, $F_{os} = 1$, but nevertheless illustrates the importance of the consideration. This relation enables the formulation of a comfort equation for conditions of 70 F: $t_a + 1.5t_s = 175$.

From this, one can compute the required air temperature in an enclosure with homogeneous surround.

By the resistance concept:

$(t_a - t_s) / (t_a - t_d) = r_i / R = r_i U$
t_d = outside temperature;
t_a = cabin air temperature;
t_s = cabin surface temperature;
r_i = resistance of inside air film;
R and U = overall resistance and overall coefficient.

Thus, for $U = 0.3$, inside film coefficient = 1.5, and $t_d = 0$ F, the inside air temperature for the same degree of comfort as would exist if both air and walls were at 70 F, is 79.5 F.

One method of reducing radiant loss is to have surfaces of low emissivity. Ideally, an occupant of a cabin with perfectly reflecting walls would be unable to lose any energy by radiation. This would be equivalent to a cabin with walls at the same temperature as the clothing surface, 81.2 F. Comfort would be obtained with an air temperature of 53.2 F.

In a new design (3) an airplane manufacturer has used a double wall construction with hot air passing in between the two surfaces so that the inside surface is at 100 F. Ventilation air then becomes purely for ventilation purposes and is delivered at 70 F. With ventilation air at 70 F, however, a wall temperature of 81.2 F would be ample, as calculated in reference No. 2.

Panel radiation as a means of overcoming radiant losses would be very effective if it were not for the adverse shape factors of usual panel installations with respect to the occupant. Because of close spacing in an airplane, panel temperatures greater than 120 F should not be used. Due to interference of passengers seated along the walls, radiant wall panels are sometimes ineffective. On the other hand, when ceiling panels are used, the shape factor is so poor that as little as 7 per cent of the energy put into the panel is received by the occupant.

Some designers have proposed panels that are designed as convective panels with the radiant heat due to elevated temperature as a sort of "bonus." The objection to this is that the panel convective system is usually not as efficient as a ventilation system and the incidental radiation is rather insignificant.

The solution would seem to be a combination system whereby the interior wall surface temperature was maintained at about 81 by passing air through some space between the exterior skin and the insulation or interior finish. The windows could be double pane construction with warm air passing in between two panes. Local cold spots should be covered with highly polished aluminum in order to cut down radiation and improve reflectivity.

Methods of Air Supply and Distribution

If a ventilation system is used for heating, by the proper use of aspirating nozzles and outlets for the vent system, good distribution and small temperature gradients throughout the airplane can be obtained. The problem requires complete and intelligent surveys by the designer.

In some pressurized cabins, air is supplied through separate duct systems. In one design, there is a cold air system, spent air system, outlet and ground conditioning system, and a hot air and supercharging system (5). The scoop for the cold air system is on the upper side of the fuselage. Air is rammed into this duct, extending over the length of the fuselage, by the motion of the plane. The duct is divided into two parts, the lower part for distribution to the cabin, the upper part for distribution to individual outlets at each seat and upper berth. Spent air is exhausted into the lower fuselage sections through vents in the passenger floor. The air is discharged from the airplane through the supercharger control units, the auxiliary outlet, or both. The ground conditioning system supplies conditioned air through the fresh air and warm air supply ducts. The warm air system is installed in duplicate.

PASSENGER comfort is of the utmost importance in airplane design, for very little time could be saved by a long distance, high speed flight of 12 hours duration if it took six hours for mental and physical recuperation. Mr. Martin discussed physiological considerations, heat transmission calculations, and "heat balance" in the June HPAC; this month, he writes on radiation loss and its effect on comfort, and methods of air supply and distribution. His article is based on his thesis submitted in partial fulfillment of the requirements for the degree of master of aeronautical engineering, College of Engineering of New York University.

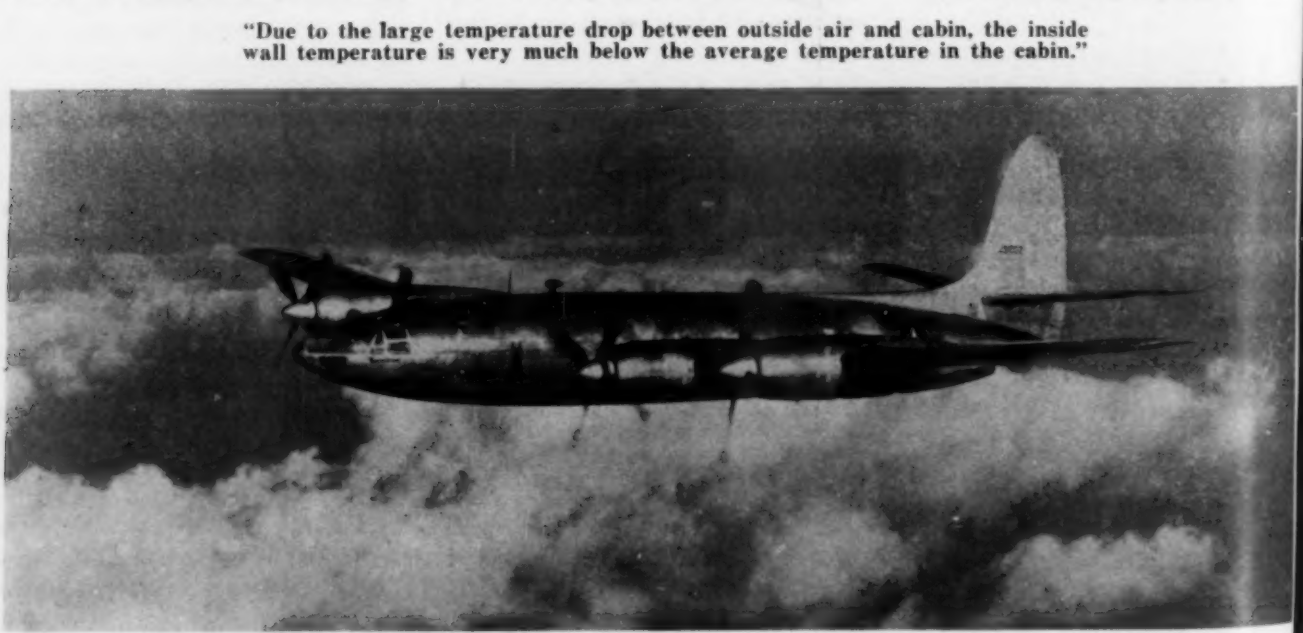
Air enters a scoop in the nose section of the inboard wing near the body and passes through a water separating device in the scoop. It then flows through ducts to the supercharger in the inboard nacelle and through a condenser, where heat is transferred to it from the steam system as required. It then enters a control unit within the body which allows it to flow, unobstructed, through the cabin for unsupercharged conditions. Outlet grilles admit air to the cabin just above the floor at each seat. The duct systems on either side of the plane are interconnected to allow for one system being inoperative. The warm air ducts are also connected to the upper fresh air duct to enable the supply of warm air to individual ventilators.

Ducting Methods

There are several basic ducting methods. Foremost among these are the overhead, sidewall, and floorboard schemes. The proper ducting system for a particular installation may be a combination of all three. The designer must consider the berthing and seating arrangements, where both might exist in the same cabin. Individual supplies, adjustable by the passenger, present a difficult problem in that the air supply is changed considerably and the distribution thrown out of balance when a number of terminals may be closed off.

Obstructions in the way of terminals will cause objectionable drafts, short circuiting of supply air to exhaust terminals will reduce the air supply, and improper distribution will cause stratification of the air. In some cases, where a large number of airplanes are involved, it would prove best, after a

"Due to the large temperature drop between outside air and cabin, the inside wall temperature is very much below the average temperature in the cabin."



complete and accurate design, to set up a complete mockup of the duct system with all obstructions and make a complete study of distribution.

In view of the large heat losses from the plane and the great amount of radiant loss from the human body, distribution of air has to be very exact. This is especially so when elevated air temperatures are used to compensate for radiant losses. A prerequisite of correct distribution is accurate calculation of the duct losses. It is very necessary to have the proper proportion of air delivered at each terminal.

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Heat Recovery Improves Diesel Engine Operation

HEAT RECOVERY is an important means of improving the operation of the present day diesel engine, according to W. C. Burgy, Caterpillar Tractor Co., in a recent talk.

There are two possible sources for recovering Btu's which are now ordinarily lost, he said—from the engine jacket cooling water and from the engine exhaust gases.

By means of a heat exchanger, the Btu's contained in the engine jacket water can be recovered, Mr. Burgy stated, and he said that with the conventional types of engines, this would represent approximately 31 per cent of all the Btu's represented in the total gallons of fuel consumed. By installing a water cooled exhaust manifold on an engine, this 31 per cent recovery can be increased by approximately 9 per cent, making the total heat recovered from the jacket water equal to around 40 per cent.

By the use of an exhaust gas heat economizer, the average efficiency of which is 55 per cent, an additional 9 per cent of the total Btu's consumed can be recovered from the exhaust gases, he reported.

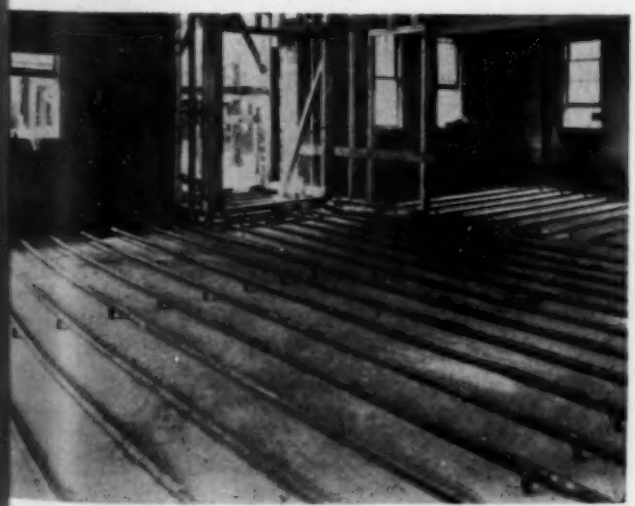
The total of these various economies amounts to 49 per cent. Add this figure to the thermal efficiency of the conventional type diesel and the overall thermal efficiency becomes 81 per cent, he said.

Nor is the above heat recovery the last word on this subject. There appears possibilities for improving this figure by a new type of jacket water cooling system known as vapor phase high temperature cooling, Mr. Burgy said.

This system consists of using a closed, pressure circuit for the jacket cooling water. By raising the pressure, the boiling point temperature of the jacket water is also raised. For instance, with a pressure of 5 lb gage on the system, the boiling point of the jacket water would be 227.1 F. At this pressure and temperature, the latent heat of evaporation is 960.4 Btu. Should process work of a plant require steam at, say, 5 lb gage, jacket water under 6 lb gage pressure could be released for process work and the result would be a gain of the latent heat, or 960.4 Btu per lb, as the water at 6 lb pressure flashed into steam at 5 lb pressure.

It is claimed that the overall thermal efficiency of one diesel plant has been increased from the normal 32 per cent to a total of 82 per cent by this method, he said.

How a radiant heating system may be installed during a remodelling project is illustrated by the American Junior Red Cross headquarters building at Erie, Pa. To fabricate the radiant heating system, G. L. Scobell Co., heating contractors, welded 1¼, 1½ and 2 in. wrought iron pipes into a grid and sinuous coils. Short lengths of 1¼ in. pipe were welded to the bottom of each coil and grid pipe to form legs. Then a gravel bed was spread on the existing concrete floor under the radiant heating element. A 3 in. concrete slab was poured on the pipes and the gravel fill, and the slab was topped with tile, applied with a mastic.



REGULATIONS REVISED

New York City Permits Use of "Freons 21, 22 and 113"

THE New York City fire department has revised the regulations governing the use of "Freon" refrigerants to permit the use of "F-21", "F-22", and "F-113". Previous to this action only "Freons 11, 12, and 114" were permitted. This revision, together with important stipulations concerning the installation and operation of these refrigeration systems, are contained in Extracts from the Administrative Code of the City of New York Relating to Refrigerating Systems.

Though the code classifies these six refrigerants as noninflammable and nonirritant, there are some 15 basic restrictions governing their use. When used in a room in which there is an open flame or apparatus to produce an open flame, the provisions of the code covering the use of irritant refrigerants apply. Systems containing more than 6 lb of "F-21" or "F-113" must be confined to separate machinery rooms which are to be maintained vaportight. For systems using more than 10 lb of any of these six "Freon" refrigerants (except class C systems located in rooms in which no open flame or apparatus to produce such flame is employed) each refrigerating machinery room must be maintained vaportight. However, in maintaining vaportightness, all doors which open to other parts of the building are required to be self-closing and close fitting and are to be kept closed at all times except during entrance or exit. All other openings which may permit the passage of vapor to other parts of the building must be vaportight and kept closed. Openings from elevator shafts are not permitted, though dumbwaiter shafts are allowed if the door openings are fitted with self-closing fire doors.

Systems employing these six refrigerants must be confined to the refrigerating machinery room except "in buildings used exclusively for ice making or for refrigerating purposes, or when not carried above the first floor in business buildings, or in the business sections of business buildings provided the entire system is confined to one floor in the space occupied by a single tenant, or in the business section of a residence building when not carried above the first floor, or in a residence building occupied by not more than two families." "Freon" systems, installed in accordance with the above limitations, may be of the direct type (wherein the refrigerant is circulated to the space refrigerated) when used for air conditioning.

It is to be noted that the requirement of a separate machinery room prevails in every case where systems employ more than 6 lb of either "F-21" or "F-113". Such systems, therefore, are practically restricted to the indirect method, if the cooling effect is to be applied outside the machinery room.

Installations in hospitals and theaters are confined to fireproof machinery rooms, used for no other purpose and in which no open flames are employed, except in certain special cases wherein class C systems using not more than 10 lb of "Freons 11, 12, 22, and 114" or not more than 6 lb of "F-21" or "F-113" may be installed as follows:

a) In diet kitchens of hospitals provided the space is cut off from the rest of the building by a tight fitting, self-closing door or doors, and provided there is at least one window to the outside air. All apparatus for producing open flames must be vented to the outside.

b) In a hospital private room provided not more than two persons are helpless and/or given medical treatment, provided there are tight partitions, and provided the door or doors are tight fitting and self-closing. In addition, no apparatus for producing an open flame is permitted.

c) In restrooms, smoking, or lounging rooms of theaters provided no open flame or apparatus for producing an open flame is employed.

Systems employing any of these six "Freon" refrigerants may be installed in rooms where there is an open flame or apparatus for producing an open flame provided that the flame and/or apparatus is equipped with a hood and an independent mechanical ventilation system so arranged as to convey all the products of combustion to the outside of the building. This provision applies only to diet kitchens in hospitals and to class C systems (those having not more than 20 lb of refrigerant) in schools, churches, dance halls, court rooms, police stations, jails, asylums, subways, passenger depots, rooms opening into passenger depots and/or subways, and such other places as are deemed necessary by the fire commissioner.

Refrigerating systems are not permitted to be installed or maintained in or on the stairways, halls, lobbies, entrances, exits, or auditoriums of any building.

Requirements as to both natural and mechanical ventilation of refrigerating machinery rooms, test pressures, the setting of safety valves, the required size of pressure relief valves, and the storage of refrigerants are included in these extracts which are published in the Vol. 31, No. 24 issue of the Bulletin of the Board of Standards and Appeals, City of New York, Municipal Bldg., New York City. Single copies may be had by mail for 8 cents.

Industry Needs a Simple Thermometer for Effective Temperature

THE EDITOR—Frankly, my reaction to the discussion by Erwin J. Davis on dry bulb thermometers in the July HPAC is that most of us in the industry long have recognized the "comfort zone" and the necessity for associating dry bulb readings with wet bulb temperatures or with relative humidity.

I agree with the first of Mr. Davis' two recommendations, at the close of his remarks, that it would be a good idea to remove dry bulb thermometers from public view in air conditioned buildings.

I do not, however, agree with the second point, wherein the author recommends the alternative of placing a wet bulb thermometer beside every dry bulb thermometer. It is perfectly true that a dry bulb and wet bulb reading determines the effective temperature, but who (in addition, possibly, to the building engineer) knows what effective temperature means?

It is believed that something, for public view, would be demanded, and we have had direct experience along that line. Consequently, I think that it behooves the industry to develop a direct reading effective temperature thermometer, which could be labelled "effective temperature" and would not require any computation on the part of the observer. About now, someone will stand up and say that there are effective temperature indicators available. That is perfectly true, but the need is for something simpler and less expensive, which could be exposed to public view in air conditioned spaces. The public could be educated to the idea of thinking along the lines of effective temperature, but it hardly could be expected to learn definitions and to become expert in the computing of dry bulb and wet bulb relationships.—J. R. VERNON, sales promotion manager, Johnson Service Co.

REPLY BY AUTHOR—I totally agree that industry should develop a simple and inexpensive thermometer which will directly read the *effective* temperature, and nothing else but effective temperature. But the point is this: We do not have any simple and inexpensive direct reading thermometers for effective temperature, and we do not know when such an instrument will be developed and placed on the market. Until that happy time arrives, it is our job to correct the present unsatisfactory system in which the dry bulb thermometer stands alone, and therefore useless, in numerous buildings.

Mr. Vernon is correct in stating that the public cannot be educated to compute relationships of dry bulb thermometers and wet bulb thermometers, but the public can be educated to do something which will serve the purpose. Drugged with heat, soaking with perspiration, one hundred million Americans will make this profound statement: "It isn't the heat; it's the humidity." And they believe that fact just as surely as they believe that Monday follows Sunday! But, how did those one hundred million Americans come to learn that fact about humidity? They were educated to it.

What do you hear in winter about humidity? Not a peep! When you complain that your office is cold, does your boss point to the dry bulb thermometer and tell you, "It isn't cold; it's the lack of humidity?" No, you won't hear about the lack of humidity, and when I say lack of humidity I, of course, am referring to the relative humidity being low.

But I believe that you will hear, before long, just as much discussion about the lack of humidity indoors, in winter, as you now hear about the excess of humidity in the summer.

A combination indoor-outdoor thermometer is now a very common article. The public has learned to read this type of double thermometer. And the public can learn to read a wet bulb and a dry bulb thermometer. Then what? We just said that the public cannot be educated to compute effective temperature. You will find, if you accept one-half of the difference between the readings of the wet bulb and of the dry bulb, and call that figure the effective temperature, you will be close enough for public use.

Educate the public and you simultaneously create a demand for improvements.—ERWIN J. DAVIS.

"OPEN FOR DISCUSSION"

STOKER INDUSTRY AIDS SMOKE ABATEMENT WORK

THE EDITOR—I would like to offer the following comments in connection with Raymond R. Tucker's comments on my paper on page 83 of the August HPAC.

The discussion of factors affecting smokeless operation of underfeed stokers in my paper does not constitute a suggested code. Although it may prove to be a useful guide in the preparation of rules and regulations for stoker installations under city smoke ordinances, its purpose, in a broader sense, is stated as follows:

". . . to further the interests of smoke abatement and public safety through a better understanding of the factors in connection with the application and installation of underfeed stokers that may effect their smokeless and safe operation."

The standards and recommended practices referred to in the discussion and published in the Technical Manual of the Stoker Manufacturers Association are based upon years of experience of the industry's leading engineers with underfeed stokers operating throughout the coal burning areas of the U. S. and Canada. Therefore, local conditions are taken into account, making the material generally suitable for application in all communities.

Depending upon the nature of a standard or recommended practice and the relative importance of maximum and minimum requirements, either or both limits may be specified. In conformity with the usual practice of specifying maximum heat release, the SMA heat release figure is a maximum recommended value.—LLOYD L. CONNELL, technical director, Stoker Manufacturers Association.

FLOOR CONSTRUCTION FOR RADIANT HEATING

THE EDITOR—In regard to the paper on Improved Floor Construction for Radiant Heating Systems, by John E. Peterson, published in the July HPAC, the author is correct, of course, in his main conclusion that heat lag occasioned by the mass of heat capacity of the earth under concrete floors creates a serious problem so far as floor panel heating systems are concerned. The paper is regarded as a useful contribution in that it emphasizes this problem and indicates the possibility of a solution—or, at least, amelioration by means of an insulating layer (slag suggested) under the concrete slab. Unfortunately, neither this nor any other work has yet presented a quantitative solution to the problem, and it may be that such a solution is impossible of attainment.

I am not so optimistic as the author about obtaining theoretical solutions to problems of this nature. The conductivities and heat capacities of the materials concerned—such as earth, concrete, etc.—are too variable, as he observes. Otherwise, I am in agreement with his conclusions. He is to be commended for pointing out that the heat lag of floors with heating systems of this kind can be very serious.—R. S. DILL, chief, heat transfer section, National Bureau of Standards.

THE EDITOR—I was greatly interested in John E. Peterson's representation of the heat loss problem of radiant heating panels imbedded in ground slabs which appeared in the July HPAC. I hope you do not mind if I ask a few questions in connection with this article.

In Fig. 1 of the article, is the distance designated by x measured downward from the horizontal panel surface which is at a temperature of 100 F? In other words, when using moist clay, is there a temperature of 60 F in this clay base at an elevation 1 ft under the floor surface? I suppose that the tests represented in Fig. 1 were taken under conditions where the edge losses of the test panel could be disregarded.

We are particularly interested in finding out the magnitude of heat losses through the building edges which are commonly used to melt the snow around the building circumference or to be dissipated by the outside air. Has Mr. Peterson made any studies as to the magnitude of such edge losses of panels installed on different types of ground? We have known of cases where the heat loss of floor panels amounts to as much as 45 per cent of the total heat input and, then again, of other cases where this heat loss could practically be ignored. Although we are aware of the fact that the panel heat loss decreases in importance with increasing size of the building, we would like to get as much information on the subject as we can.—R. G. VANDERWEIL, project engineer, development department, Chase Brass & Copper Co.

REPLY BY AUTHOR—I am glad to learn of Mr. Vanderweil's interest in my article and will try to answer the questions presented.

The distance x is measured downward from the surface of the fill; that is, from the underside of the slab panel. The mathematical analysis used with this figure requires that the entire fill area is presumed to be of uniform temperature and infinite thickness. The curves are calculated on the basis of suddenly applying and maintaining 100 F at the top of the fill. This is not the surface temperature of the panel.

These curves are purely a mathematical calculation, but since they are based on physical characteristics of the material and these characteristics are commonly used in heat transmission problems, it appears satisfactory to use this approach as a comparative analysis of materials.

Our investigations indicated that the fill area stores large quantities of heat units and that this storage, which eventually becomes lost, is in addition to the edge loss. If the floor slab is continuous from inside to outside, there is considerable horizontal heat flow. Ordinarily the exterior edge of the slab is above grade, so this flow is dissipated to the atmosphere. In one case we found this edge loss to be in the order of 10,000 Btu per hr at design conditions. The need for insulating the slab at the perimeter has been pointed out by other investigators.

WHEN ENGINEERS gather to hear a paper or talk, there comes a time when the meeting is thrown "open for discussion." We follow this custom in these columns, which provide an opportunity to comment on articles published; discuss other topics of interest to heating, piping, and air conditioning engineers and contractors; ask questions; make suggestions, etc.

I believe the downward flow will vary considerably from job to job, depending upon the soil conditions in the locality and no doubt they vary with seasons. For this reason it appears prudent to establish a fixed condition by providing a waterproof barrier and fill of known value so that the downward heat flow is not only reduced but is also known.

I agree that, presumably, as a building becomes larger these factors may reduce, but our investigations were conducted in residences and here losses of considerable magnitude are possible. In the case referred to above, we found 25,000 Btu per hr downward flow over a 10 hr period. This, together with the edge loss (approximately 10,000 Btu per hr) amounted to about 50 per cent of the capacity of the furnace.—JOHN E. PETERSON, research engineer, Minneapolis-Honeywell Regulator Co.

DUCT SIZING WITH PARTIAL STATIC REGAIN

THE EDITOR—I have read the article and data sheet by R. H. Anderegg and F. W. Hutchinson on Duct Sizing with Partial Static Regain, published in the August HPAC, and find them most interesting. The chart should prove to be very useful to the duct designer and I believe the assumptions made in the derivations of the equations are reasonable in the light of available information.

It is possible the chart that has been prepared by Messrs. Anderegg and Hutchinson may need to be revised when more experimental data are available.

If the paper is reprinted, I would appreciate having a copy for my files.—J. R. FELLOWS, professor of mechanical engineering, University of Illinois.

NEW BOOKS & REPORTS

Pipe Thread Standards Revised

The first data which appeared as a complete pipe thread standard was published in the Transactions of the American Society of Mechanical Engineers in 1919. This material was later approved by the American Standards Association as the first "American Standard". The first revision was published in 1942 by the ASME as ASA B2.1-1942. Due to the need for additional data on pipe threads and gaging during World War II, a further revision was undertaken which resulted in the present standard, B2.1-1945.

This 1945 revision provides the required information on the subject of pipe threads for any reasonable project requiring some form of thread. The symbols covered in Appendix I, while practically the same as those in the 1942 edition, were reviewed in connection with the nomenclature project of the conference on unification of engineering standards, held in Ottawa, 1945. These symbols are now considered standard by all groups in this country.

The standard now includes the fundamentals for the "Dryseal Pressure-Tight Joints". The close requirements for this type of pipe thread were developed for service where the use of any lubricant or sealer is considered undesirable. The details for the crests and roots, which cover the only difference between this and the regular thread, are illustrated. Some of the subjects covered in the standard either by description, illustration, data in tabular form, or in the appendixes are:

Form of thread and basic formulas
Limits on crest and root of external and internal taper pipe threads
Specifications for plug and ring gages for taper pipe threads
Method of gaging and manufacturing tolerances for threaded product
Basic dimensions, American Standard taper pipe threads
Dimensions of threaded plug and ring gages
Specifications for "Dryseal Pressure-Tight Joints"
Specifications for rail fitting joints
Specifications for straight pipe threads, mechanical joints, and locknut connections
Definitions and letter symbols
U. S. Army Ordnance method of gaging pipe threads
U. S. Army-Navy-Aeronautical special gaging practice
API method of gaging line pipe
Method of gaging "Dryseal" taper and straight pipe threads
Twist drill diameters for tapped holes for pipe threads
Threading of pipe for American Standard steel flanges
Locknut pipe joint for tank nipples
Small hose coupling screw threads
Taper pipe threads connecting discharge valves to compressed gas cylinders
Rigid steel conduit and fitting threads
Straight pipe threads for bung plugs and flanges of steel barrels and drums
Table of pitch diameters of taper pipe threads

The American Standard on Pipe Threads, B2.1-1945, was prepared by the sectional committee on standardization of pipe threads, sponsored by the American Gas Association and the ASME. Copies are now available from the ASA headquarters, 70 E. 45th St., New York 17, N. Y. The price per copy is $1.50.

Recommendations Given for Preventing Corrosion in Stoker Feed Screws

Corrosion and failure of the feed screw in small underfeed stokers of the residential type has been a matter of concern to both stoker manufacturers and retailers. Many reasons have been advanced as the cause of corrosion in stoker screws. These include (1) heating of the retort end of the screw, (2) moisture on the coal, (3) backward flow of the products of combustion from the retort, (4) acidity of the coal, and (5) the presence of calcium chloride on the coal.

This problem has been the subject of a study carried on at the fuel research laboratory, Battelle Memorial Institute under the sponsorship of Bituminous Coal Research, Inc., and the Calcium Chloride Association. The findings and recommendations are contained in technical report No. 9, The Corrosion of Feed Screws of Small Underfeed Stokers, by R. A. Sherman, J. F. Foster, and D. A. Hinckle.

Laboratory tests described in the report indicate that the conditions necessary for corrosion are (1) evaporation of moisture from the coal at the retort end of the screw, (2) backward flow of water vapor in the feed tube toward the hopper, and (3) condensation of the moisture at a cooler section of the screw. According to the authors, if these conditions are present, the presence of natural acidity in the coal or of calcium chloride will tend to accelerate the rate of corrosion. If the three essential conditions listed above are not present, the corrosion will not be severe, even if the coal is acid or is treated with calcium chloride. To prevent or reduce corrosion, it is recommended that heating of the coal at the retort end and the backward flow of gases be prevented.

Copies of this 23 page report may be obtained from Bituminous Coal Research, Inc., 912 Oliver Bldg., Pittsburgh 22, Pa. Copies are priced at 25c each.

Welding Manual on Aluminum Published

Gas welding, arc welding, resistance welding, brazing and soldering are the principal headings in a new 88 page manual devoted to the latest practices and recommendations for joining aluminum by these processes.

After a brief discussion of the more widely used aluminum alloys from the standpoint of factors affecting their weldability, the book analyzes recommended gas welding practice with sections covering such topics as edge preparation, cleaning and preheating, the welding flame, welding rods, fluxes, and technique.

The arc welding material includes chapters on the metal arc, carbon arc, atomic hydrogen, and inert-gas-shielded processes. Resistance welding is broken down into sections on spot, seam, and flash welding, with information on preweld cleaning, recommended machine settings, tip pickup, and where to look to correct trouble. Subsequent chapters discuss furnace and torch brazing, soldering, inspection of welds, metallurgy of welds, and the like.

The book is available from Reynolds Metals Co., Dept. 47, 2500 S. 3rd St., Louisville 1, Ky. The price is $1.00 each.

MEETINGS & CONVENTIONS

Technical Sessions and Inspection Trips Planned for ASME Fall Meeting

The 1946 fall meeting of the American Society of Mechanical Engineers, 29 W. 39th St., New York City, will be held at the Statler hotel, Boston, September 30 to October 3. Eighteen technical sessions are planned, among which are several on heat transfer, hydraulics, management, power, and applied mechanics. Inspection trips will include visits to the Massachusetts Institute of Technology, General Electric Co., and a number of other places of interest.

Gas Industry Exhibit and AGA Meeting to be Held Concurrently

Many new developments in the gas appliance, equipment, and supply field will be presented for the first time publicly at the national exhibition of the Gas Appliance Manufacturers Association, to be held October 7 to 11 in Atlantic City and in conjunction with the annual convention of the American Gas Association.

More than 125 manufacturers have already signed up for exhibition space and more than 10,000 gas utility executives, members of the liquefied petroleum gas industry, and others are expected to attend the AGA convention, according to L. C. Harvey, president of the Gas Appliance Manufacturers Association, 60 E. 42nd St., New York 17, N. Y.

All-Industry Refrigeration Show to Have Varied Exhibits

Opening of the 4th All-Industry Refrigeration and Air Conditioning Exposition, which will be held in the Cleveland Auditorium, October 29 to November 1, is expected to reveal more than 300 exhibits of the industry's newest equipment and latest developments.

The exposition, the first since January 1941, is sponsored by the Refrigeration Equipment Manufacturers Association, comprising more than 100 member companies, with the cooperation of the Frozen Food Locker Manufacturers and Suppliers Association, whose members now number more than 7000 throughout the country. In the frozen food field alone there will be many exhibits of home, farm, and commercial freezers, storage cabinets, and lockers, together with associated equipment. Specially designed equipment for locker plants, retail stores and frozen food packers will be on display.

Industrial applications of mechanical refrigeration such as the shrink fitting of metal parts, treatment of steel, and the testing of various products will be represented among the exhibits. Air conditioning equipment designed for residential, office, commercial, and industrial use will also be on display.

During the week of the show, associations of wholesalers, service engineers, contractors, and frozen food locker operators will hold meetings. The show will be open to the public from 7 to 10 p.m., Thursday, October 31.

Exposition headquarters are at 1107 Clark Bldg., Pittsburgh, Pa. R. K. Hanson, executive secretary of REMA, is show director.

IHF Plans 11th Annual Meeting

A series of three or four round table conferences is being arranged in connection with the 1946 annual meeting of member companies of the Industrial Hygiene Foundation. The general meeting will be held at the Mellon Institute, Pittsburgh, the foundation's headquarters, on Thursday, November 7. As in past years the program will deal with broad subjects of managemental interest respecting the well-being of men at work.

Subjects of more specialized interest will be discussed in informal conferences to be held on the day before and the day after the general meeting. A meeting for physicians from member companies and a conference for plant engineers are tentatively planned for November 6.

Plan Public Health Meeting

The 74th annual meeting of the American Public Health Association, 1790 Broadway, New York City, will be held in Cleveland, November 11 to 14.

Annual Meeting of AWS and Metal Congress in Atlantic City

The 1946 annual meeting of the American Welding Society will be held during the National Metal Congress and Exposition in Atlantic City, November 17 to 22.

The Atlantic City convention hall will be the scene of the exposition and the various meetings of the AWS will be held at the Ambassador hotel.

All persons planning to attend and who do not have hotel reservations are urged to write to the Housing Bureau, 16 Central Pier, Atlantic City, indicating first, second, and third choice of hotels.

National Power Show and ASME Annual Meeting in December

The 17th National Exposition of Power and Mechanical Engineering and the annual meeting of the American Society of Mechanical Engineers will be held in New York City during the first week in December.

Many innovations in power equipment and exhibits of scientific interest will be on display at the power show. A model of the first gas turbine actually proposed for power generation in the utility field will be shown. Other features will include exhibits of power generation equipment, equipment for power transmission to machines, and some phases of power utilization.

The show is being conducted by the International Exposition Co., with permanent headquarters in Grand Central Palace, where the show will be held. ASME headquarters are at 29 W. 39th St., New York City.

ASHVE Annual Meeting, Heating and Ventilating Show to Be in Cleveland

The 53rd annual meeting of the American Society of Heating and Ventilating Engineers will be held in Cleveland, the last week in January, 1947. The society is sponsoring the 7th International Heating and Ventilating Exposition which will be held concurrently in that city.

The exposition is being managed by the International Exposition Co., Grand Central Palace, New York City. ASHVE headquarters are at 51 Madison Ave., New York 10, N. Y.

QUESTION OF THE MONTH

How Design Snow Melting Coils?

P. G., a reader of Heating, Piping & Air Conditioning, noticed the statement in a recent article that "it is the usual practice to design snow melting coils for a snowfall of 1 in. per hr." He therefore asked these questions, which were published last month:

1) *How many Btu per hr is 1 in. per hr of snow?*

2) *Are these rates different for different parts of the U. S. A.? Where are data available?*

3) *What efficiency is assumed in transferring inches of snow per hour into coal consumption?*

Following are some answers to his questions:

DEPENDS ON MANY FACTORS

I assume that "P. G." is seeking general information, because the question of snow melting is somewhat involved since it depends on so many factors.

As regards his first question, it can be assumed that the average weight for 1 in. of snow is approximately 0.5 lb. This, however, depends on the temperature at the time snow is falling. On this basis it can be assumed that the Btu required will be 72 per sq ft of area. To this must be added the heat loss to the atmosphere and the heat loss downward, both of which depend on the type of construction.

For instance, if the pipes are buried below a concrete layer 4 in. or 6 in. thick, it will take a higher temperature to penetrate the concrete than it would where the pipes are buried under a comparatively thin layer of asphalt, as would be the case with a residential driveway. The amount of heat passing downward would depend on the soil and road bed below the pipes, and this can vary from 10 to 25 per cent.

As regards the second question, the rate of snowfall in different parts of the U. S. A. does, of course, vary, but the weight of the snow is of great importance. However, since the weight of snow is greater when the air temperature is higher, there is to a certain extent a compensating effect on the heat loss, although more heat is required for temperatures in the neighborhood of 32 F than for very low temperatures. In the northern parts of the U. S. A. it is safer to assume 2 in. per hr of snowfall, although in some cases heavier falls have been recorded.

The third question of course depends on the construction, as mentioned in my answer to the first question. It also depends on the amount of snowfall per hour. For an average snowfall of 1 in. per hr, the fuel consumption would probably be 9 lb of coal per sq yd of area over a 24 hr period of snowfall.

I hope the above will give "P. G." sufficient information to meet his requirements.—T. N. A.

EXPERIENCE WITH 30 JOBS

It is impossible to give "P. G." a specific and definite answer to the questions he outlind.

As yet, there are no specific data which can generally be applied to all snow melting jobs. The 30-some jobs with which I am familiar are being studied every winter and before long more definite information should be available.

In general, it might be said that the best results so far have been obtained by installing 1¼ in. pipe on 18 in. centers in the entire concrete slab from which the snow is to be removed. This will take care of any snowfall up to about 4 in. per hr and the usual practice is to design snow melting systems for such a rate of snowfall.

Referring to "P. G."s first question, theoretically it takes 144 Btu to melt 1 lb of snow or ice. The weight per cubic foot varies according to outside temperature conditions.

Regarding the second question, it is of course obvious that because of the rates of snowfall and because of the prevailing temperatures during those snowfalls, the rate at which heat is supplied to melt this snow would be different in different parts of the U. S. However, the above figures have proved safe for any part.

The answer to the third question is dependent on a great many things, as we must know the type of coal being used, how the coal is fired and what the overall efficiency of the boiler plant is. A rough answer to the question is that 50 per cent of the Btu content of the coal can be transferred into snow melting.—E. M.

$64 QUESTION

"P. G." asks a "$64 question" on snow melting coils.

The local weather bureau office estimates that 10 in. of snow fall is equal to 1 in. of precipitation.

The temperature of the snow will, in all probability, be equal to or lower than the temperature of the air.

I find the specific heat of ice given as 0.1988 at –200 C and 0.53 at –10 C, and 1.0 at the freezing point.

So to estimate the heat required, find out from the local weather bureau how much snow equals an inch of precipitation and the rate of snowfall per hour, from which you can determine the number of pounds to melt.

Multiply this by the degrees below freezing of the air plus 144 (latent heat of snow) to get the total Btu required.

Then comes the rub. Where does the snow go from here? Is there a nice unfrozen, steam heated gutter to carry it away, or will it have to be evaporated, which, of course, will add another 1000 Btu per lb to the load, plus 180 Btu?

As to efficiency, I would guess at, say, 40 per cent where you think you have a gutter to carry the water away to 80 per cent where you have to evaporate the snow.

The above, of course, are estimates or guesses and are not based upon any observations. They must, therefore, be taken with a grain of salt—but don't mix the salt with the snow.—E. W.

WHAT AMOUNT OF HEAT FOR TWO DRY KILNS?

YOU ARE invited to submit an answer to the following question from "H. W." for publication in these pages. Suitable answers will be paid for at regular rates, with additional payment for sketches. You are invited, too, to submit a question about heating, piping, or air conditioning for publication here.

A plant making veneer crates and hampers for the fruit and vegetable trade has two dry kilns each 7 x 22 x 120 ft. Eight thousand wet crates are put through these per day. Each crate carries about 8 lb of water. Banks of steam coils at one end with powerful exhaust fans at the other furnish the present drying medium. No one knows the capacity of the fans, but they are extremely powerful.

With the exception of the wire stitching department, this plant is shut down on Saturday. Because of this, they have to dry material during the week for this day's run. This adds to the cost and gives poorer work. These cannot lay wet because of mildew.

This firm is interested in getting some auxiliary heat to enable them to operate the kilns on Saturday. It is my feeling that a direct fired oil heater would be the most economical. As they normally fire the boilers with mill waste, of which they do not have enough and have to add oil, I feel that they would be better off to use oil heat all of the time as the direct application would be more economical than burning it under the boilers, with the characteristic saw mill losses in between.

I have been trying to locate manufacturers of units suitable for this job and perhaps you can help me locate some. One that I contacted claims that this load would take 6,400,000 Btu per hr. Another company states it would require 280,000 Btu per hr with electrical unit heaters. With this difference, it is very apparent that someone is "all wet."

Can the necessary data as to the amount of heat required be supplied? It will certainly be appreciated.—H. W.

"SCRAP FOR PEACE"

"Everyone who is waiting for steel can help the steel industry by assisting the movement of scrap," says the committee on iron and steel scrap of the American Iron and Steel Institute. The committee gives the following reasons for the scrap shortage:

During the coal strike a larger percentage of scrap than normal was used in furnace charges because there was no coal to make pig iron. Consequently, already-low supplies of scrap were brought to rock-bottom. Strikes in consuming and fabricating industries, from which a large part of the scrap normally comes, have cut that supply drastically. Another factor is that battlefield scrap has not been returned in expected quantities due to chaotic conditions in the countries abroad. Also, the uncertain price situation has hampered flow of scrap.

Unable to get new equipment, potential buyers have been forced to hold onto their old—railroads, for instance, cannot tear up old track or scrap locomotives until replacements are available. Farmers cannot scrap old equipment until they get new equipment. Auto "graveyard" scrap is low, too, because such yards are a source of valuable parts and this scrap will not move as scrap until the automobile demand is saturated. Also, there is a labor shortage in scrap yards.

Scrap normally makes up 50 per cent of the charge in an open hearth furnace, from which by far the greatest proportion of all steel comes. The biggest producers of scrap are generally the railroads, the automotive industry, farms, and factories.

The steel industry was able to increase its operating rate in July by using more pig iron in place of scrap. However, the limit has just about been reached in this respect.

Twenty-five to 30 open hearth furnaces were idle August 1 for lack of scrap, against 12 idle for that reason July 1. Meanwhile, during July receipts of scrap by steel mills fell from an average of 60 per cent of requirements to approximately 40 per cent of requirements.

DATA SHEET

THREE-SHIFT SCHEDULE

By M. W. GARLAND, Chief Engineer, Refrigerating Machinery Dept., Frick Company, Waynesboro, Pa.

Three-Shift Schedule for a Plant Having a Chief Engineer and Three Watch Engineers, Each Serving 48 Hours per Week Only

Week	Shift	Sunday	Monday	Tuesday	Wednesday	Thursday	Friday	Saturday
No. 1	8-4	A	A	A	A-E	A-E	A-E	E
	4-12	B	B	E	B	B	B	B
	12-8	C	E	C	C	C	C	C
No. 2	8-4	E	B	B	B-E	B-E	B-E	B
	4-12	C	C	E	C	C	C	C
	12-8	A	E	A	A	A	A	A
No. 3	8-4	E	C	C	C-E	C-E	C-E	C
	4-12	A	A	E	A	A	A	A
	12-8	B	E	B	B	B	B	B
No. 4	8-4	E	A	A	A-E	A-E	A-E	A
	4-12	B	B	E	B	B	B	B
	12-8	C	C	C	C	C	C	E

At the end of each week, each operator advances to the second shift ahead. Example: When Operator A has completed four weeks, one week under each of the four outlined above, he then goes back to the plan of Week No. 1 on the 12-8 shift.

A=Watch Operator. B=Watch Operator. C=Watch Operator. E=Chief Engineer.

Three-Shift 48-Hour Schedule for Engineers

THE schedule on the preceding page was developed to meet the needs of plants of small and medium size, where it is necessary to maintain three shifts of watch engineers. The plan as shown covers the shifts to be worked by a chief engineer having three watch operators under him, but would be equally applicable if there were more operators on each trick. The schedule is based on experience; it was worked out by M. W. Garland, of the Frick Company.

The shifts are so arranged that each man will have a Saturday and a Sunday off, at one time or another, every month. Each man also has a Monday and a Tuesday off.

No one works more than 48 hours a week. The chief works six days a week, the same as the watch operators. A man who prefers to work the same shift can be "frozen" on that shift without upsetting the convenience of the other men or the effectiveness of the system. The day off of the frozen man will, however, change each week.

The shifts are changed at midnight—this gives both the men on the night turns some daylight work. The 4 - 12 shift is even preferred by some men to regular day work, as it still leaves most of the night for sleeping and allows half a day of free time for other activities. The 12 - 8 man likes his hours better than the old hours of 8 p.m. to 4 a.m., which never gave him a free evening.

Under this plan the chief engineer has a watch engineer to help in repair and maintenance work three days a week, during daylight hours, when stores and service men, if needed, are doing business. This means that all four men always work Wednesdays, Thursdays and Fridays.

At the end of each week, each of the watch engineers advances two shifts ahead. The "A" man who starts on the all-daylight shift of 8 to 4 skips the 4-to-12 shift his second week and goes on the 12-to-8 turn. The third week he works from 4 to 12 p. m. The fourth week he is back on the 8-to-4 all-daylight shift. With the beginning of a new month he moves two shifts ahead again, and takes the place of "C" on the schedule.

Heating, Piping & Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

Energy Losses at Suction Hoods

By Allen D. Brandt* and Russell J. Steffy**

This paper is the result of research carried on by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, at its Research Laboratory, 7218 Euclid Ave., Cleveland 3, Ohio in cooperation with the U. S. Public Health Service.

SUMMARY—The energy losses at the entrance to 175 different types of suction openings were determined and the coefficients of entry were calculated. Effects of the following factors on the coefficient of entry were studied: Distance of suction measurement downstream from hood throat; duct velocity; degree of hood taper (included angle); hood shape; hood face area to throat area ratio; flanges; and size of hood throat.

Purpose of Study

THE energy loss at suction hoods is an important consideration in the design of local exhaust systems for industrial atmospheric sanitation.[1-4] Since this loss can be minimized by proper hood design, it has been the tendency among designers in recent years to shape the hoods in such manner that this loss would be as small as practically possible. On the basis of existing data, this procedure necessitated long transformation sections from the hoods or hood faces to the ducts and frequently made the design needlessly difficult owing to the interference with parts of the equipment or machine.

Recent field observations indicated that the energy loss at hoods is probably not so high as indicated in the literature. If these observations are correct, it would not be necessary to use long transformation sections and, as a result, hood design would be less difficult and less material would be needed for hood construction.

In addition, the static suction in a duct close to an exhaust hood frequently affords the most convenient, if not the only practicable method of estimating the quantity of air being exhausted from a given operation.[5] This information is essential to industrial hygiene engineers since the degree of contamination control is a function of the quantity of air exhausted. It is also extremely important that the relationship between the exhaust ventilation rate and the contaminant concentration in the atmosphere be known for different hoods and different operations if it is to serve as a guide on other similar installations. If, as indicated by field observations, most energy losses at hoods reported in the literature[1-3] are too high, an avoidable error is introduced in the ventilation rate calculation.

Consequently, it was deemed advisable to study the energy loss at a large variety of hoods to determine as accurately as practicable the true values and to provide information which would serve to estimate with a fair degree of accuracy the coefficient of entry for any hood.

Experimental Procedure

The experimental procedure and a discussion of the types of hoods investigated in this study are covered in detail in Appendix A and will be summarized only very briefly at this point.

A total of 175 different hoods, including (1) different shaped hoods, (2) different tapers or included angles, (3) different throat sizes, (4) different face area to throat area ratios, (5) flanged and unflanged and (6) different flange sizes, were tested at four different rates of air flow. The experimental setup is shown in Fig. 1, which is

*Sanitary Engineer, U. S. Public Health Service, assigned to ASHVE Research Laboratory. Member ASHVE.
**Research Assistant, ASHVE Research Laboratory.
[1]Exponent numerals refer to Bibliography.
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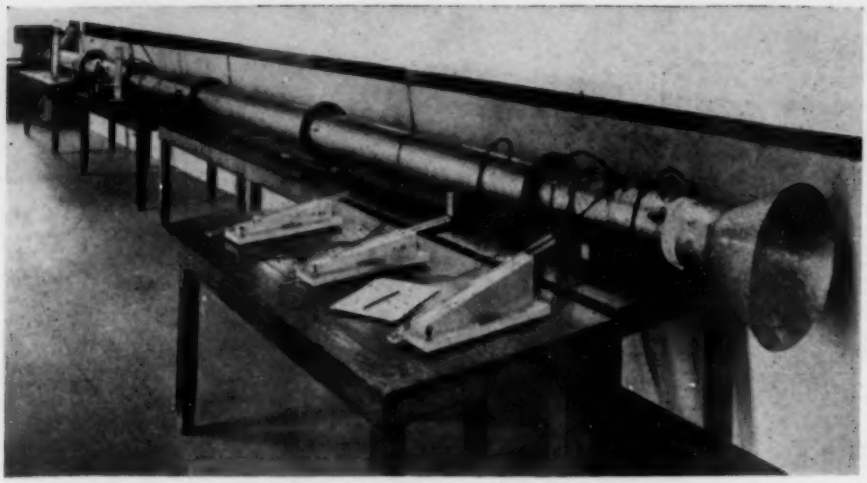

Fig. 1. Setup for testing suction hoods

self-explanatory (see Appendix A for more detail).

Findings

The coefficients of entry for all hoods were computed by

$$C = \frac{Q}{4008A\sqrt{h_s}} \quad \ldots\ldots\ldots\ldots (1)$$

where C = coefficient of entry
Q = measured quantity of air being exhausted
A = area of the duct in which h_s was determined
h_s = measured static suction in inches of water

This is the same as the more common expression which follows since $Q = 4008A\sqrt{h_v}$

$$f = \sqrt{\frac{h_v}{h_s}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

where f = coefficient of entry
h_v = duct velocity pressure in inches of water
h_s = static suction in duct near hood in inches of water

Not all authors agree on what is meant by the *coefficient of entry*. To industrial hygiene engineers the coefficient of entry expresses the ratio of the actual air flow into a hood to the theoretical air flow as indicated in the foregoing equations. One author,[9] on the other hand, defines the coefficient of entry as the ratio of the average velocity over the full area to the actual velocity in the vena contracta. Since there is considerable static regain downstream from the vena contracta, the coefficients reported by this author are very low. For the purposes of design of local exhaust systems and measurement of air quantities exhausted from given operations, the ratio of the actual to the theoretical air flow rate has much more value than other expressions. The terms *energy loss* or *entrance loss* and *coefficient of entry* are used more or less interchangeably throughout this paper. This may be confusing to some readers unless it is remembered that the energy loss (resistance or pressure drop) is a function of the coefficient of entry squared as shown in Equation (3).

$$h_e = \left(\frac{1 - C^2}{C^2}\right) h_v \quad \ldots\ldots\ldots\ldots (3)$$

where h_e = energy loss in inches of water
C = coefficient of entry
h_v = duct velocity pressure in inches of water

In Table 1 are summarized the coefficients of entry for the 167 hoods having 8 in. diameter throats. The hoods listed include round, rectangular, square, flanged and unflanged types. In addition to these hoods the investigation included a few having different diameter throats but which were otherwise proportional. These will be discussed separately later. The hood identification used in this and other

Table 1—Summary of Test Results for Hoods Having 8 in. Throat Diameters
(*Duct velocity* 4000 *fpm*)

Hood Number	Shape	Included Angle* Deg	Coefficient	Hood Number	Shape	Included Angle* Deg	Coefficient*	Hood Number	Shape	Included Angle* Deg	Coefficient
1-2.5	Round	10	0.926	8-4	Square	40	0.944	14-5F-1	Rect.	107	0.883
1-2.5F	Round	10	0.967	8-3	Square	40	0.942	15-5	Round	136	0.882
1-2	Round	10	0.922	8-3F	Square	40	0.950	15-5F	Round	136	0.895
1-1.6	Round	10	0.888	8-3F-8	Square	40	0.955	15-4	Round	136	0.892
1-1.2	Round	10	0.823	8-3F-4	Square	40	0.952	15-3	Round	136	0.895
2-2.5	Rect.	17	0.933	8-3F-2	Square	40	0.950	15-3F	Round	136	0.891
2-2.5F	Rect.	17	0.968	8-2	Square	40	0.933	15-2	Round	136	0.902
2-2	Rect.	17	0.925	8-2F	Square	40	0.938	15-2F	Round	136	0.886
2-1.6	Rect.	17	0.903	8-2F-8	Square	40	0.939	16-5	Round	90	0.948
3-6	Square	90	0.901	8-2F-4	Square	40	0.941	16-5F	Round	90	0.958
3-6F	Square	90	0.908	8-2F-2	Square	40	0.941	16-4	Round	90	0.952
3-5	Square	90	0.895	8-2F-1	Square	40	0.942	16-3	Round	90	0.950
3-4	Square	90	0.894	9-5	Rect.	29	0.969	16-3F	Round	90	0.955
3-3	Square	90	0.898	9-5F	Rect.	29	0.981	16-2	Round	90	0.952
3-3F	Square	90	0.900	9-4	Rect.	29	0.962	16-2F	Round	90	0.947
3-2	Square	90	0.887	9-3	Rect.	29	0.952	16-1.6	Round	90	0.962
3-2F	Square	90	0.902	9-3F	Rect.	29	0.966	16-1.4	Round	90	0.957
4-5	Rect.	109	0.870	9-2	Rect.	29	0.938	16-1.2	Round	90	0.811
4-5F	Rect.	109	0.872	9-2F	Rect.	29	0.960	16-F-½R	Round	90	0.839
4-4	Rect.	109	0.867	10-5	Rect.	81	0.936	17-5	Round	21	0.967
4-3	Rect.	109	0.858	10-5F	Rect.	81	0.952	17-5F	Round	21	0.983
4-3F	Rect.	109	0.873	10-4	Rect.	81	0.939	17-4	Round	21	0.963
4-3F-4	Rect.	109	0.876	10-3	Rect.	81	0.926	17-3	Round	21	0.960
4-3F-2	Rect.	109	0.880	10-3F	Rect.	81	0.943	17-3F	Round	21	0.975
4-2	Rect.	109	0.845	10-2	Rect.	81	0.912	17-2	Round	21	0.946
4-2F	Rect.	109	0.879	10-2F	Rect.	81	0.950	17-2F	Round	21	0.964
5-5	Rect.	134	0.860	11-5	Round	58	0.971	18-1	Round	0	0.717
5-5F	Rect.	134	0.883	11-5F	Round	58	0.978	18-F-10	Round	180	0.803
5-4	Rect.	134	0.862	11-5F-8	Round	58	0.979	18-F-8	Round	180	0.819
5-3	Rect.	134	0.855	11-5F-4	Round	58	0.974	18-F-4	Round	180	0.814
5-3F	Rect.	134	0.886	11-5F-2	Round	58	0.973	18-F-2	Round	180	0.818
5-3F-8	Rect.	134	0.888	11-5F-1	Round	58	0.974	18-F-1	Round	180	0.835
5-3F-4	Rect.	134	0.884	11-5F-1R	Round	58	0.978	18-F-1R	Round	180	0.832
5-3F-2	Rect.	134	0.890	12-6	Square	58	0.910	19-1	Square	0	0.697
5-2	Rect.	134	0.858	12-6F	Square	58	0.923	19-F	Square	180	0.791
5-2F	Rect.	134	0.892	12-6F-8	Square	58	0.922	19-F-8	Square	180	0.795
5-2F-8	Rect.	134	0.895	12-6F-4	Square	58	0.923	19-F-4	Square	180	0.792
5-2F-4	Rect.	134	0.892	12-6F-2	Square	58	0.924	19-F-2	Square	180	0.795
5-2F-2	Rect.	134	0.894	12-5	Square	58	0.919	19-F-1	Square	180	0.800
5-2F-1	Rect.	134	0.888	12-4	Square	58	0.922	20-5	Round	50	0.980
6-5	Rect.	54	0.950	12-3	Square	58	0.916	20-4	Round	50	0.978
6-5F	Rect.	54	0.960	12-2	Square	58	0.895	20-3	Round	50	0.973
6-5F-8	Rect.	54	0.949	12-2F-2	Square	58	0.905	20-2	Round	50	0.960
6-5F-4	Rect.	54	0.953	12-2F-1	Square	58	0.905	21-5	Round	30	0.975
6-5F-2	Rect.	54	0.956	13-5	Rect.	77	0.917	21-4	Round	30	0.975
6-5F-1	Rect.	54	0.954	13-5F	Rect.	77	0.918	21-3	Round	30	0.970
7-5	Round	40	0.985	13-4	Rect.	77	0.906	21-2	Round	30	0.955
7-5F	Round	40	0.987	13-3	Rect.	77	0.893	22-5	Round	115	0.944
7-4	Round	40	0.976	13-3F	Rect.	77	0.918	22-4	Round	115	0.944
7-3	Round	40	0.968	13-2	Rect.	77	0.893	22-3	Round	115	0.945
7-3F	Round	40	0.978	13-2F	Rect.	77	0.922	22-2	Round	115	0.944
7-2	Round	40	0.958	14-5	Rect.	107	0.870	23-5	Square	115	0.890
7-2F	Round	40	0.962	14-5F	Rect.	107	0.878	23-4	Square	115	0.890
8-6	Square	40	0.940	14-5F-8	Rect.	107	0.882	23-3	Square	115	0.890
8-6F	Square	40	0.955	14-5F-4	Rect.	107	0.880	23-2	Square	115	0.884
8-5	Square	40	0.950	14-5F-2	Rect.	107	0.878				

*Major angle only for rectangular hoods

Fig. 2. Some of hood types studied. Small ratio of hood face area to throat area

tables is significant as explained more fully in Appendix A. The number (or numbers) before the first dash identifies the hood group, the one following the dash the approximate hood face area to throat area ratio. F indicates a flanged hood with 12 in. wide flanges unless otherwise specified, the number following the second dash indicates the flange width in inches if other than 12 in., and the R indicates a circular flange (most flanges on circular hoods were square).

A study of Table 1 will show that only 17 of the hoods tested have coefficients of entry below 0.850. Of these, 14 (groups 18, 19 and 16-F-½R) represent some variety of an open duct end and 2 (1-1.2 and 16-1.2) have unusually low face area to throat area ratios. The majority of the coefficients (62 per cent) are 0.90 or above and of a total of 82 unflanged hoods studied 26, or 32 per cent, had coefficients of 0.950 or higher. It is apparent that, with but few exceptions, the results obtained in this study are higher in general than those commonly reported in the literature. Notable among the exceptions are the values for flanged and unflanged duct ends (hood groups 18, 19 and 16-F-½R) which check very well with the usual values of 0.82 and 0.72, respectively. The results on hood group 21 (average about 0.97) agree fairly well with data on a similar hood given in reference 9 (0.944) but are considerably higher than results reported elsewhere, such as in reference 5, where a value of about 0.875 is given.

The fact that most of the results obtained in this study are substantially higher than those in common use caused considerable concern and led to check and recheck of the test procedure from all angles. As all attempts in this direction confirmed the results listed, the authors are at a loss to explain the lower values reported elsewhere. As one check on test accuracy, a contracting jet orifice, borrowed from Case School of Applied Science, having, according to their studies, a coefficient of 0.985, was tested. Several tests on this orifice at a duct velocity of about 4000 fpm gave an average value of 0.989, which is a satisfactory check and seems to substantiate the results reported in this paper. It was suggested by one other worker[10] that the roughness of the taper to throat connection is extremely important and will influence the value of the coefficient a great deal; the smoother or rounder this joint, the higher will be the coefficient of entry. That the nature of this connection will have a significant influence on the energy loss at a hood seems apparent, yet a careful study of the construction of most of the hoods used in this study indicates the presence of two features which should place them in the high energy loss group. These features are: (1) the tapered section terminated inside the rolled edge of the throat section thereby presenting a straight and sharp edge at the point where the change in direction of air movement takes place and (2) at the rolled edge of some of the throat sections the pipe is drawn to form an internal bead the inside diameter of which is significantly less than the pipe diameter (see hoods 4, 7 and 12 in Fig. 2). In spite of these apparent bad features in construction and the two different types of construction discussed in Appendix A, the measured coefficients fall in line with each other, as will be demonstrated more clearly later.

Effect of Distance from Throat on Results

As is shown in Appendix A, readings were taken at a number of distances downstream from the throat. This was done purposely to determine how far from the throat readings should be taken to get reliable results; that is, how far down the duct influence of the vena contracta would be felt sufficiently to give high or erratic results. This is a very important point in field work where the static reading is used to estimate the quantity of air being exhausted through a hood since the ductwork downstream from the hood is frequently so crooked, inaccessible or damaged as to make it imperative that the reading be taken very close to the hood.

The results indicate that one pipe diameter is adequate for most hoods but that a greater distance is desirable if the included angle of the transformation section or hood is great. Of the 23 hood groups, having 8 in. diameter throats, studied, only seven were found to have the lowest static suction reading more than one pipe diameter downstream from the throat. Pertinent data on six of these hood groups are summarized in Table 2.

Table 2—Unflanged Hoods Having Highest Pressure (Lowest Suction) More Than One Pipe Diameter Downstream From Hood Throat

Hood Group	Shape	Major Included Angle Deg	Suction at One Pipe Diameter In. Water	Lowest Suction In. Water	Distance from Hood Throat in Pipe Diameters
4	Rectangular	109	1.378	1.356	1¾
5	Rectangular	134	1.361	1.349	1¾
13	Rectangular	77	1.262	1.258	1¾
15	Round	136	1.263	1.254	1¾
18	Round	0[a]	2.363	1.926	4½+
18-F	Round	180[b]	1.584	1.506	4½+

[a]Round unflanged open duct end.
[b]Flanged round duct end.

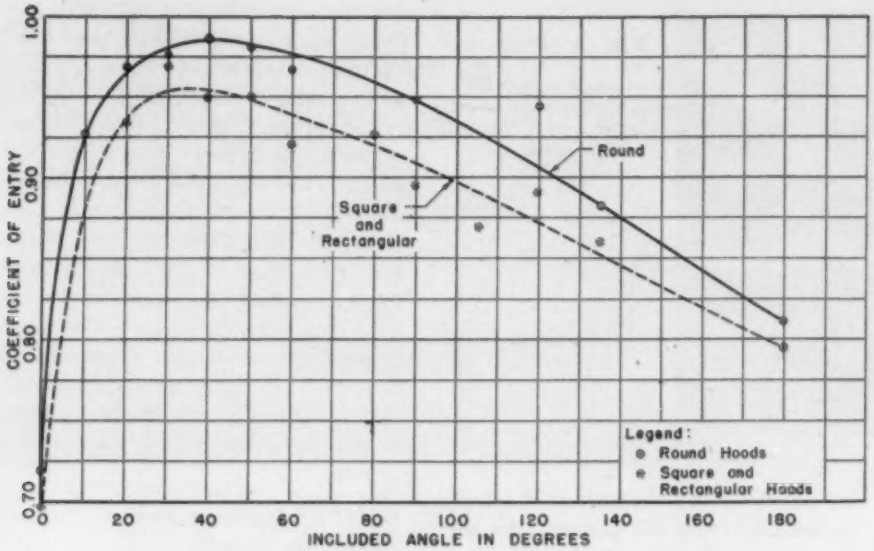

Fig. 3. Effect of the included angle in unflanged hoods on the coefficient of entry

It is not apparent why hood group 13 is among this number. Other rectangular and square hoods having greater included angles have not given similar results. Furthermore, this same characteristic did not appear for all sizes of the hood group nor at all duct velocities. It is believed, therefore, that this hood group is among the lot by virtue of a small experimental error (note the small difference) or some unusual characteristic of the air stream. For open duct ends or flanged duct ends, however, the static reading should be taken not less than three pipe diameters (preferably 5) downstream from the throat. The results in Tables 1 and 2 indicate that even though the lowest static reading for some hoods with severe taper (over 100 deg included angle) exists more than one pipe diameter downstream from the throat, for all practical purposes one pipe diameter is satisfactory up to included angles of 180 deg (flanged duct ends). For included angles greater than 180 deg and for flanged and unflanged duct ends, the reading should be taken at a point not less than 3 pipe diameters downstream from the throat; the suction beyond 3 pipe diameters decreases only very little.

Effect of Static Tap Location on Coefficient

The static suction close to the throat of a rectangular to round taper or rectangular hood is not the same at all locations around the throat, particularly if the sides ratio of the hood is quite large. Readings taken near the throat are considerably higher in the area of the major included angle than they are in the area of the smaller angle. Results indicate, however, that even with a sides ratio of 4 to 1 and with major and minor included angles of 134 deg and 0 deg, respectively, the readings at one pipe diameter downstream from the throat vary only little at different locations around the duct as regards the relation to the different included angles. There is, of course, considerably more turbulence at this short distance from the throat with a hood of this type than with a round or square hood and suction readings should be taken at three or four points around the duct, if possible, to obtain the best results.

Effect of Duct Velocity on Coefficient of Entry

The coefficients of entry were determined at duct velocities of about 2000, 3000, 4000 and 5000 fpm on 70 different unflanged hoods. The total averages of these coefficients at the different velocities are 0.925, 0.920, 0.922 and 0.922, respectively. Hence it is obvious that the duct velocity within the limits of 2000 to 5000 fpm has no significant influence on the coefficient of entry.

Effect of Hood Taper on Entrance Loss

The unflanged hoods studied included a large variety of different tapers. In Table 3 are summarized the coefficients for hoods of different included angles, all other variables having been kept constant, except as noted. It is apparent from the table that the best hood from the energy loss standpoint has an included angle of about 40 deg. This is true whether the hood is round, square or rectangular. The approximate coefficients for unflanged, round, square and rectangular hoods having throat diameters of 8 in. and face area to throat area ratios of 5, and for included angles of 0 to 180 deg are shown in Fig. 3. Of interest are the coefficients listed in Table 3 for hoods 16, 17, 20 and 21 (included angles of 90, 20, 50 and 30 deg respectively) since the construction of these hoods was such that the taper to throat section junction appeared to be considerably smoother than the other hoods (see Appendix A). Yet the coefficients are not out of line with those of the other

Table 3—Effect of the Included Angle in Unflanged Hoods on the Coefficient of Entry

(*Duct velocity about* 4000 *fpm.* — $\frac{A_1}{A_2}$ *about* 5 *except where otherwise stated*)

Hood Type	Included Angle in Degrees												
	0[a]	10	20	30	40	50	60	80	90	105	120	135	180
	Coefficients of Entry												
Round	0.717	0.926[c]	0.967	0.975	0.985	0.980	0.966		0.948		0.944	0.883	0.810
Square	0.697				0.950		0.919		0.895		0.890		0.798
Rectangular[b]	...		0.933[c]	0.969		0.950		0.927	0.870			0.860	

[a] Open end duct—same as 360 deg.
[b] Major angles only for rectangular hoods.
[c] $\frac{A_1}{A_2} = 2.5$

hoods. Both Table 3 and Fig. 3 indicate that in general the coefficients for square and rectangular hoods are in the order of 0.03 lower than for similar round hoods, this difference being greater in the middle section of the curve than at the extremities.

These data are at considerable variance with other information in the literature on this same subject. One source[6] suggests that the lowest energy loss is accomplished with a hood having an included angle of 13 deg. Another source[5] suggests that a hood having an included angle of 60 deg or more has the same coefficient of entry as a flanged duct end. As indicated in Table 3, a 60 deg hood is an excellent one and the coefficient is in the order of 0.94 not 0.82 as for a flanged duct end. It must be noted, of course, that these values are for hoods having 8 in. diameter throats. For different size hoods the coefficients will differ as shown later.

Table 4—Effect of Hood Face Area to Throat Area Ratio on Coefficient of Entry for Unflanged Hoods (*Duct velocity* 4000 *fpm*)

Hood Group	Shape	Included Angle[a] Deg	Coefficients for Different Area Ratios 6	5	4	3	2	1.6	1.2
1	Round	10				0.926	0.922	0.888	0.823
2	Rectangular	17				0.933	0.925	0.903	
3	Square	90	0.901	0.895	0.894	0.898	0.887		
4	Rectangular	109		0.870	0.867	0.858	0.845		
5	Rectangular	134		0.860	0.862	0.855	0.858		
7	Round	40		0.985	0.976	0.968	0.958		
8	Square	40	0.940	0.950	0.944	0.942	0.933		
9	Rectangular	29		0.969	0.962	0.952	0.938		
10	Rectangular	81		0.936	0.939	0.926	0.912		
12	Square	58	0.910	0.919	0.922	0.916	0.895		
13	Rectangular	77		0.917	0.906	0.893	0.893		
15	Round	136		0.883	0.893	0.895	0.902		
16	Round	90		0.948	0.952	0.950	0.952	0.959	0.811
17	Round	21		0.967	0.963	0.960	0.946		
20	Round	50		0.986	0.978	0.973	0.960		
21	Round	30		0.975	0.975	0.970	0.955		
22	Round	115		0.944	0.944	0.945	0.944		
23	Square	115		0.890	0.890	0.890	0.884		

[a]Major angle only for rectangular hoods.

Table 5—Effect of Flanges on Coefficient of Entry (*Duct velocity* 4000 *fpm, flange width about* 12 *in.*)

Hood Number	Shape	Included Angle Deg	Coefficient Unflanged	Flanged	Hood Number	Shape	Included Angle Deg	Coefficient Unflanged	Flanged
1-2.5	Round	10	0.926	0.967	9-2	Rectangular	29	0.938	0.960
2-2.5	Rectangular	17	0.933	0.968	10-5	Rectangular	81	0.936	0.952
3-6	Square	90	0.901	0.908	10-3	Rectangular	81	0.926	0.943
3-3	Square	90	0.898	0.900	10-2	Rectangular	81	0.912	0.950
3-2	Square	90	0.887	0.902	11-5	Round	58	0.971	0.978
4-5	Rectangular	109	0.870	0.872	12-6	Square	58	0.910	0.923
4-3	Rectangular	109	0.858	0.873	13-5	Rectangular	77	0.917	0.918
4-2	Rectangular	109	0.945	0.879	13-3	Rectangular	77	0.893	0.918
5-5	Rectangular	134	0.860	0.883	13-2	Rectangular	77	0.893	0.922
5-3	Rectangular	134	0.855	0.886	14-5	Rectangular	107	0.870	0.878
5-2	Rectangular	134	0.858	0.892	15-5	Round	136	0.883	0.895
6-5	Rectangular	54	0.950	0.960	15-3	Round	136	0.895	0.891
7-5	Round	40	0.985	0.987	15-2	Round	136	0.902	0.885
7-3	Round	40	0.968	0.978	16-5	Round	90	0.948	0.955
7-2	Round	40	0.958	0.962	16-3	Round	90	0.950	0.955
8-6	Square	40	0.940	0.955	16-2	Round	90	0.952	0.947
8-3	Square	40	0.942	0.950	17-5	Round	21	0.967	0.983
8-2	Square	40	0.933	0.938	17-3	Round	21	0.960	0.975
9-5	Rectangular	29	0.969	0.981	17-2	Round	21	0.946	0.964
9-3	Rectangular	29	0.952	0.966					

Table 6—Effect of Flange Width on Coefficient of Entry (*Duct velocity* 4000 *fpm*)

Hood Number	Shape	Included Angle[a] Deg.	Coefficients for Different Flange Widths 12 in.	8 in.	4 in.	2 in.	1 in.
4-3	Rectangular	109	0.873		0.876	0.880	
5-3	Rectangular	134	0.886	0.888	0.884	0.890	
5-2	Rectangular	134	0.892	0.895	0.892	0.894	0.888
6-5	Rectangular	54	0.960	0.949	0.953	0.956	0.954
8-3	Square	40	0.950	0.955	0.952	0.950	
8-2	Square	40	0.938	0.939	0.941	0.941	0.942
11-5	Round	58	0.978	0.979	0.974	0.973	0.974
12-6	Square	58	0.923	0.922	0.923	0.924	
14-5	Rectangular	107	0.878	0.882	0.880	0.878	0.883
18 F	Round	0	0.803[b]	0.811	0.814	0.818	0.835
19 F	Square	0	0.791	0.795	0.792	0.795	0.800

[a]Major angle only for rectangular hoods.
[b]10 in. flange.

Effect of Face Area to Throat Area Ratio on Entrance Loss

A total of 16 different hood groups were studied at four different hood face area to throat area ratios. These included round, square and rectangular hoods. In addition a few hoods were studied at different ratios. The results of these tests are tabulated in Table 4.

It is apparent from these data that the face area to throat area ratio in the range of 5 to 2 has only a very small influence on the coefficient of entry; the coefficient decreasing as the ratio decreases for most hoods. The averages for all hood groups 3 to 23 inclusive are 0.931, 0.929, 0.924 and 0.916 for face area to throat area ratios of 5, 4, 3 and 2, respectively. It is interesting to note, however, that most of this reduction takes place among the hoods having a small degree of taper. Thus the average coefficients at the four different face area to throat area ratios for the seven hood groups having included angles of 90 deg or more are 0.899, 0.900, 0.899 and 0.896 as against the values given for all hood groups and 0.955, 0.952, 0.944 and 0.932, respectively, for the hoods having included angles of less than 90 deg. The ratio value of 2, however, appears to be the lower limit of the range in which the coefficients are essentially constant. Of the three hoods studied at ratios below 2, only one remained essentially the same at a ratio of 1.6 and both hoods investigated at a ratio of 1.2 showed coefficients essentially the same as for flanged duct ends.

These data indicate that, contrary to common belief[5], the hood face area to throat area ratio has very little effect on the coefficient of entry except for values less than 2. The reason for this is not apparent at the time of this writing. At first glance it would seem that the face area to throat area should have an influence on the coefficient

Table 7—Effect of Throat Diameter on Coefficient of Entry

Throat Diameter Inches	Coefficient for Different Included Angles			
	0° or 360° (open duct)	40° $\left(\frac{A_1}{A_2}=5\right)$	90° $\left(\frac{A_1}{A_2}=5\right)$	180° (Flanged Duct)
4	0.702	0.926	0.919	0.793
8	0.717	0.985	0.948	0.810
12	0.695	0.962	0.952	0.802

since the loss at the hood face obviously is greater with a small than with a large area and it seems unlikely that this difference is made up by the greater length of the larger area hood. It is hoped that this problem may be investigated further to find the reason for these results.

Effect of Flanges on Coefficients

Of the tapered hoods investigated, 39 were studied with 12 in. wide flanges and without flanges. These results, summarized in Table 5, indicate a significant but unimportant increase in the coefficients with flanges. The average of all coefficients without flanges is 0.919 and with flanges is 0.933. All hoods except three (15-3, 15-2 and 16-2) showed increases with flanges. The exceptions are the round hoods with the greatest included angles and with small face area to throat area ratios. The reasons for these decreases are very likely the same as those which caused the coefficients of these same hoods to increase as the face area to throat area decreased (see Table 4). Since the average increase is 0.014 and the median increase is 0.013, it may be concluded that in general the coefficient of a hood is increased about 0.015 by providing a 12 in. wide flange. For all practical purposes, however, this increase is too small to have significance.

Effect of Flange Width on the Coefficient

Flange widths of 12, 8, 4, 2 and 1 in. were used on some of the hoods investigated. The results of these tests are given in Table 6. It is obvious from these data that the flange width in the range studied has no significant effect upon the coefficient of entry. The average value for the coefficients of the seven hoods which were studied over the entire range of flange widths are 0.891, 0.893, 0.892, 0.894 and 0.897 for 12, 8, 4, 2 and 1 in. widths, respectively. On the basis of the data in Tables 5 and 6 it is logical to conclude that flanges have no significant effect on the coefficient of entry of hoods other than duct ends. Even a ½ in. wide flange (see hood 16-F-½R in Table 1) has about the same coefficient as wider flanges (see 18-F in Table 6).

Effect of Large Adjacent Surfaces on Coefficient

A few tests were run on different hoods with large flat surfaces adjacent to one, two, or three sides of the hoods, the surfaces being parallel to the center line of the exhaust duct. These conditions were intended to simulate those existing in industry when a hood is laid on, or located adjacent to, a table top and possibly one or two vertical partitions are located at the ends or sides of the hood to eliminate cross drafts or for other purposes. Even though the number of tests was very small, the results indicate that such conditions have no important influence on the coefficient of entry.

Effect of Hood Throat Size on Coefficient

Since for similar hoods the ratios of the hood face perimeter and the throat perimeter to the hood face area and the throat area, respectively, decrease as the hood or throat size increases, it seemed logical that the coefficient of entry would be affected by the throat size for hoods of similar shape and proportions. It was believed that the coefficient would increase as the size increases and vice versa. Four com-

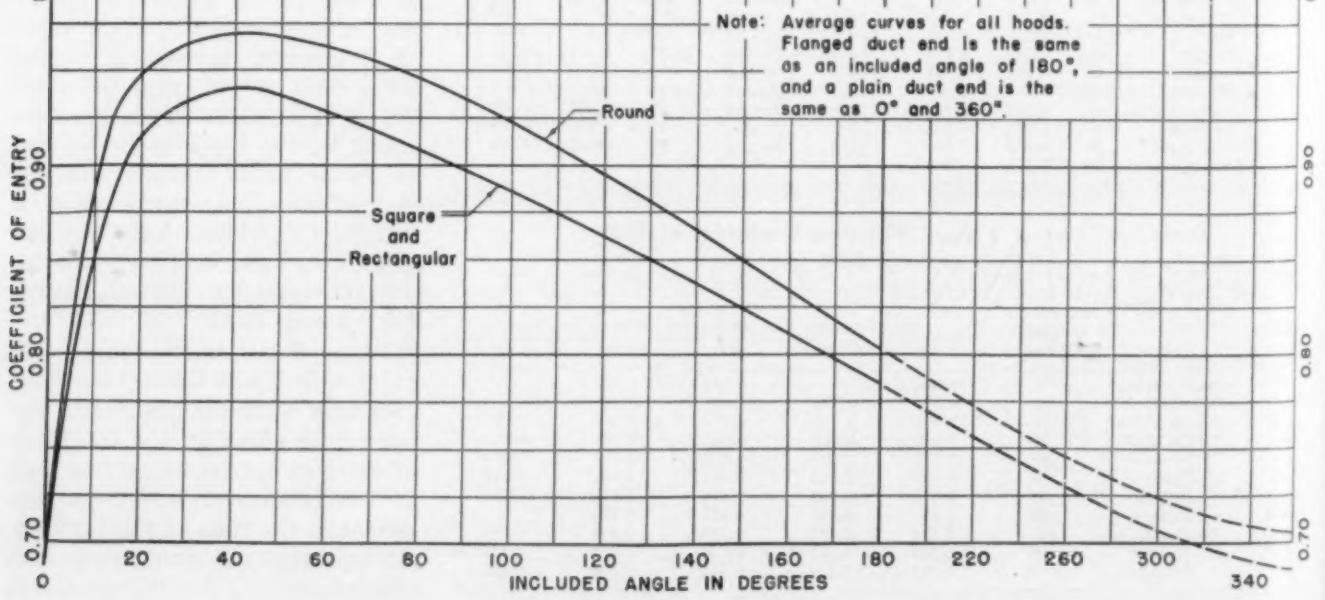

Fig. 4. Coefficients of entry for hoods under suction

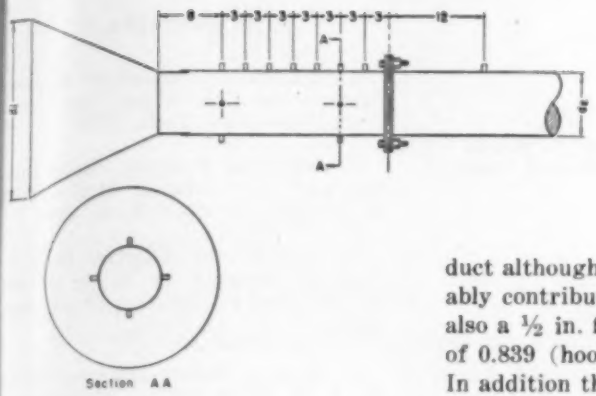

Fig. 5. Duct section showing static taps

parable hoods of each 4 in. and 12 in. throat size were studied and the results are summarized in Table 7. These results are very interesting, since the larger size hoods did not show the anticipated lower energy losses and the coefficients of the smaller hoods were lower than expected. It is not clear to us why the coefficients of the larger hoods are lower than those of the hoods having 8 in. diameter throats. Variations of the order shown in the table for the 40 and 90 deg tapered hoods having 8 in. and 12 in. throats might be explained on the basis of minor differences in construction; a point which was discussed earlier. Other factors which may have contributed to these very small inconsistencies, particularly in the case of the plain and flanged duct ends, are: (1) inaccurate measurement of the inside pipe area, (2) roundness of open end, (3) thickness of metal and (4) type of construction. The inside diameters of the ducts were determined by measuring the diameter at three or four places and averaging these measurements. Since these values were found to be close to 8 in. and 12 in. respectively, these figures were used in the calculations and might easily have varied from the exact values sufficiently to have caused the small inconsistencies noted. The 8 in. duct was made of 16 gage metal and was definitely more nearly round than the 4 in. or 12 in. ducts. The greater thickness of the metal in the 8 in. duct, also, would tend to increase the coefficient since it served as a very narrow flange. This small difference may appear at first thought to be of no consequence yet actual tests on an 8 in. duct of 20 gage material gave a result 0.011 lower than that for the 16 gage duct although other variables probably contributed to this difference; also a ½ in. flange had a coefficient of 0.839 (hood 16-F-½R, Table 1). In addition the smooth seam of the 8 in. duct contrasted to the usual construction of the 12 in. duct may have added a small amount to the higher coefficient of the 8 in. duct. These several factors may well account for the fact that the 8 in. coefficient (average 0.865) was slightly higher than the 12 in. coefficient (average 0.853) rather than vice versa. This difference however, is too small to have any practical significance. The difference between the coefficients for the 4 in. hoods (average 0.825) and the 8 in. hoods is sufficiently large to warrant consideration even in field work if accuracy of results is important. However, the greatest deviation of any single value in Table 7 from the average for all three sizes of similar hoods is less than 4 per cent. This is well within the range of permissible error for most field work.

Conclusion

The energy losses at the entrance to 175 different types of suction openings were determined and the coefficients of entry were calculated. Effects of the following factors on the coefficient of entry were studied:

1. Distance of suction measurement downstream from hood throat
2. Duct velocity
3. Degree of hood taper (included angle)
4. Hood shape
5. Hood face area to throat area ratio
6. Flanges
7. Size of hood throat

On the basis of the data obtained in this study it may be concluded that:

1. Reliable results will be obtained if suction readings are taken one or more pipe diameters downstream from the throat for hoods having included angles up to 180 deg and three or more pipe diameters downstream from the throat for included angles of 180 deg (flanged pipe end) to 360 deg, and for 0 deg (plain pipe end).
2. Duct velocity in the range of 2000 to 5000 fpm has no significant effect on the coefficient of entry.
3. The degree of hood taper (included angle) is the most significant factor affecting the coefficient of entry (see Fig. 3). The coefficients vary from about 0.70 for 0 deg included angle (plain pipe end) to a maximum of about 0.98 for a round hood with an included angle of 40 deg.
4. The hood shape, whether round, square or rectangular, has a significant, though relatively unimportant, effect on the coefficient of entry. The coefficients for round hoods are in the order of 0.02 to 0.04 higher than for square and rectangular hoods. The differences are greater in the middle

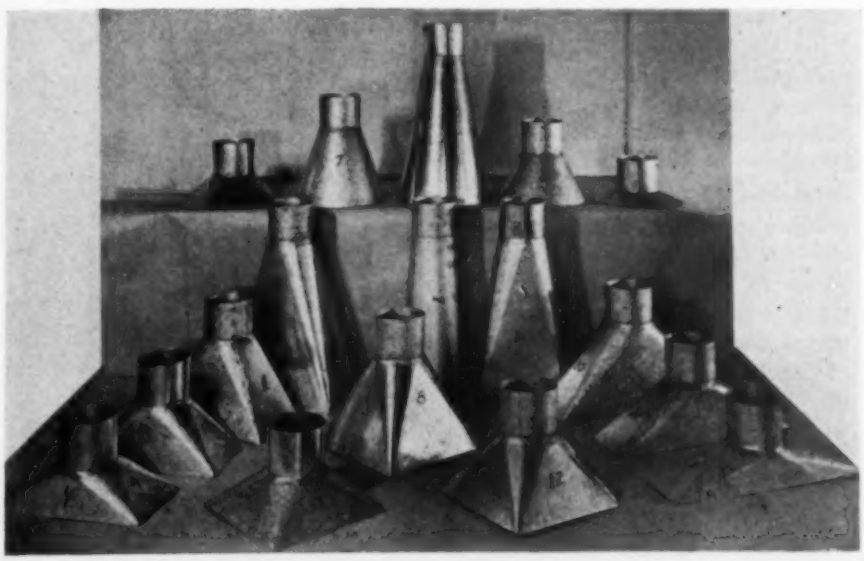

Fig. 6. Some of hood types studied. Large ratio of hood face area to throat area

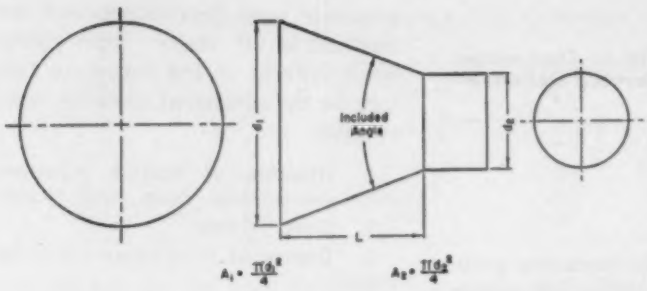

Fig. 7. Round unflanged hoods

portion of the curve than at the ends (see Fig. 3).

5. The hood face area to throat area ratio in the range of five to two has no important effect upon the coefficient of entry. Below 2 this ratio becomes of concern, because the coefficient approaches that of a flanged duct end which it equals at a ratio of 1.2.

6. Flanges have no important influence on the coefficients of tapered hoods. They do, however, have a very important influence on the coefficients of plain duct ends (0 deg included angle).

7. The size of the hood throat has a significant, though not too important effect on the coefficient of entry for similar hoods. An 8 in. diameter throat gave higher results than 4 in. or 12 in. diameter throats, although the difference between the 8 in. and 12 in. results is not significant.

8. Entry coefficients accurate to within about plus or minus 5 per cent may be predicted if the hood shape and included angle are known. Curves for this purpose are shown in Fig. 4. These curves differ slightly from those in Fig. 3 since they have been modified to take into account the effect of the hood throat sizes or duct sizes in the range commonly encountered in local exhaust systems.

Most of the results obtained in this study differ considerably from many of those reported in the literature and in common use today. With the exception of plain duct ends and flanged duct ends the results shown are higher than most of those cited in the literature. All attempts to find the reason for these differences were unsuccessful.

When considering the data and statements in this paper, it must be borne in mind that reference is made only to the coefficient of entrance, not to the relative merits of one hood to another as regards its performance. For example, while a flange does not improve a hood in regard to entrance loss, it improves the air flow pattern in front of the hood a great deal; the wider the flange the better, in this respect. Likewise, the hood face area, while unimportant from the viewpoint of entrance losses, has a decided influence on the air flow pattern in front of the hood. Consequently, the statements in this paper must not be construed as implying that one hood is no better than another in regard to the air flow pattern produced by it.

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Table 8—Unflanged Round Hoods Studied
(*Dimensions and Data*)

	(Refer to Fig. 7 for Letter Designations)						
Hood Number	d_1 In.	d_2 In.	A_1 Sq In.	A_2 Sq In.	$\frac{A_1}{A_2}$	L In.	Included Angle Deg
1-2.5	12¾	8	128	50	2.6	27½	10
1-2	11	8	95	50	1.9	18	10
1-1.6	10	8	79	50	1.6	11½	10
1-1.4	9½	8	71	50	1.4	8⅜	10
1-1.2	8¾	8	60	50	1.2	4¼	10
7-5	18	8	254	50	5.1	13¾	40½
7-4	16	8	202	50	4.0	11	40½
7-3	13¾	8	149	50	3.0	8	40½
7-2	11¼	8	99	50	2.0	4½	40½
11-5	18	8	254	50	5.1	9	58
15-5	18	8	254	50	5.1	2	136½
15-4	16	8	202	50	4.0	1½	136½
15-3	13¾	8	149	50	3.0	1¼	136½
15-2	11¼	8	99	50	2.0	⅛	136½
16-5	18	8	254	50	5.1	5	90
16-4	16	8	202	50	4.0	4	90
16-3	13¾	8	149	50	3.0	3	90
16-2	11¼	8	99	50	2.0	1⅛	90
16-1.6	10¼	8	82	50	1.6	1	90
16-1.4	9½	8	71	50	1.4	¾	90
16-1.2	8¾	8	60	50	1.2	½	90
17-5	18	8	254	50	5.1	27	21
17-4	16	8	202	50	4.0	21¼	21
17-3	13¾	8	149	50	3.0	15¾	21
17-2	11¼	8	99	50	2.0	8¾	21
18-1	8	8	50	50	1.0	—	0
20-5	17¾	8	247	50	4.9	11	50
20-4	16	8	202	50	4.0	9	50
20-3	14	8	154	50	3.1	6¾	50
20-2	11¼	8	99	50	2.0	3¾	50
21-5	18	8	254	50	5.1	19	30
21-4	16	8	202	50	4.0	15¼	30
21-3	14	8	154	50	3.1	11½	30
21-2	11¼	8	99	50	2.0	6¼	30
22-5	18	8	254	50	5.1	3¼	115
22-4	16	8	202	50	4.0	2½	115
22-3	13¾	8	149	50	3.0	2	115
22-2	11⅛	8	104	50	2.1	1	115
24-5	9	4	64	13	4.9	6½	40
25-5	9	4	64	13	4.9	2½	90
26-1	4	4	13	13	1.0		0
27-5	27	12	572	113	5.1	7½	90
28-5	27	12	572	113	5.1	21	40
29-1	12	12	113	113	1.0		0

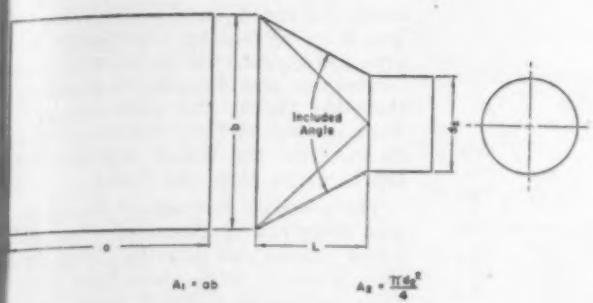

Fig. 8. Square unflanged hoods

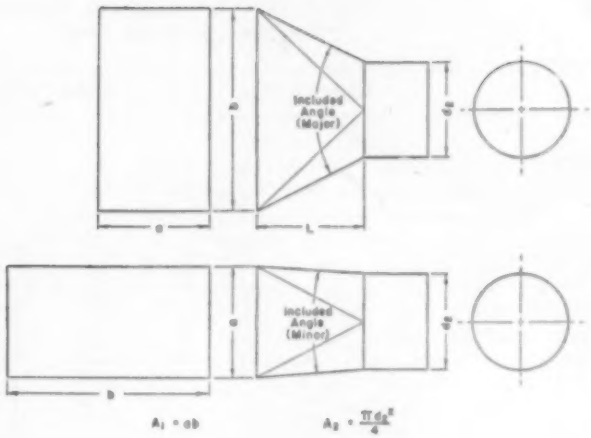

Fig. 9. Rectangular unflanged hoods

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Table 9—Unflanged Square Hoods Studied
(*Dimensions and Data*)

	(Refer to Fig. 8 for letter designations)							
Hood Number	a In.	b In.	d_2 In.	A_1 (a × b) Sq In.	A_2 Sq In.	$\frac{A_1}{A_2}$	L In.	Included Angle Deg
3-6	18	18	8	324	50	6.5	5	90
3-5	16	16	8	256	50	5.1	4	90
3-4	14	14	8	196	50	3.9	3	90
3-3	12	12	8	144	50	2.9	2	90
3-2	10	10	8	100	50	2.0	1	90
8-6	18	18	8	324	50	6.5	13¾	40½
8-5	16½	16½	8	272	50	5.4	11¾	40½
8-4	14	14	8	196	50	3.9	8¼	40½
8-3	12	12	8	144	50	2.9	5½	40½
8-2	10	10	8	100	50	2.0	2¾	40½
12-6	18	18	8	324	50	6.5	9	58
12-5	16¼	16¼	8	264	50	5.3	7½	58
12-4	14½	14½	8	210	50	4.2	5¾	58
12-3	11½	11½	8	132	50	2.6	3¼	58
12-2	10	10	8	100	50	2.0	1¾	58
19-1	8	8	8[a]	64	64	1.0		0
23-5	16	16	8	256	50	5.1	2½	115
23-4	14	14	8	196	50	3.9	2	115
23-3	12	12	8	144	50	2.9	1¼	115
23-2	10	10	8	100	50	2.0	½	115

[a]Square.

Table 10—Unflanged Rectangular Hood Studied
(*Dimensions and Data*)

	(Refer to Fig. 9 for letter designations)									
Hood Number	a In.	b In.	$\frac{b}{a}$	d_2 In.	A_1 (a × b) Sq In.	A_2 Sq In.	$\frac{A_1}{A_2}$	L In.	Included Angle Deg Major	Included Angle Deg Minor
2-2.5	8	16	2.0	8	128	50	2.6	27	17	0
2-2	8	12	1.5	8	96	50	1.9	13½	17	0
2-1.6	8	10	1.3	8	80	50	1.6	6¾	17	0
4-5	11	22	2.0	8	242	50	4.8	5	109	33½
4-4	10¼	19½	1.9	8	200	50	4.0	4	109	33½
4-3	9½	15½	1.6	8	147	50	2.9	2¾	109	33½
4-2	8¾	11¼	1.3	8	98	50	2.0	1¼	109	33½
5-5	8	31½	3.9	8	252	50	5.0	5	134	0
5-4	8	24¼	3.0	8	194	50	3.9	3¾	134	0
5-3	8	19	2.4	8	152	50	3.0	2½	134	0
5-2	8	12	1.5	8	96	50	1.9	1	134	0
6-5	11	22	2.0	8	242	50	4.8	13¾	54	12½
9-5	11	22	2.0	8	242	50	4.8	27¼	29	6½
9-4	10¼	18¼	1.8	8	183	50	3.7	20	29	6½
9-3	9¼	15	1.6	8	139	50	2.8	13½	29	6½
9-2	8½	11¼	1.3	8	96	50	1.9	6¼	29	6½
10-5	8	31½	3.9	8	252	50	5.0	13¾	81	0
10-4	8	23¾	3.0	8	190	50	3.8	9¼	81	0
10-3	8	18	2.3	8	144	50	2.9	5¾	81	0
10-2	8	12	1.5	8	96	50	1.9	2¼	81	0
13-5	11	22	2.0	8	242	50	4.8	8¾	77½	20
13-4	10¼	19½	1.9	8	200	50	4.0	7¼	77½	20
13-3	9¼	15	1.6	8	139	50	2.8	4¼	77½	20
13-2	8½	11¼	1.3	8	96	50	1.9	1¾	77½	20
14-5	8	31½	3.9	8	252	50	5.0	8¾	107	0

Appendix A

Experimental Procedure

Most of the hoods studied had 8 in. diameter throats since the available experimental setup had 8 in. diameter ductwork and metering devices. The test arrangement with a round hood attached is shown in Fig. 1. The hood throats were slightly less than 8 in. in diameter to accomplish a snug inside lap fit. An air seal between the hood throats and duct was accomplished by means of adhesive tape as shown in the figure. The metering device was a 6.00 in. sharp-edged orifice located about 26 ft (39 duct diameters) downstream from the hood. The computed calibration curve for the orifice checked very closely with rates of air flow determined by pitot traverses in the duct. The variations of the computed rate of air flow from that determined by pitot tube studies by different individuals at different rates of air flow and on different days were all less than plus or minus 3 per cent.

The end of the duct to which the hoods were attached for study was a carefully made, butt seamed, almost truly round, 24 in. long, section of pipe of 16 gage material. The ¼ in. thick flange which was welded to the pipe at the downstream end served also to maintain the shape of the pipe almost truly round under all conditions. Static taps having 1/32 in. holes through the pipe wall were located in a straight line and three inches on center as shown in Fig. 5. The end taps were each 3 in. from the ends of the pipe. In addition to the single taps at the 3 and 18 in. locations three other taps were located around the pipe at 90 deg intervals (see Figs. 1 and 5) permitting average static readings around the pipe at these two locations. An additional tap was located 12 in. beyond the flange or 36 in. from the open end of the pipe as shown in the figures. The length of the throat section of all

Table 11—Flanged Hoods Studied
(Dimensions and Data)

Hood Number	d_1 In.	a In.	b In.	A_1 Sq In.	l In.	u In.	x In.	y In.	A_2 Sq In.	$\frac{A_2}{A_1}$
					Refer to Figs. 7, 8, 9 and 10 for Letter Designations)					
1-2.5F	12¾			128	42	42	14½	14½	1764	13.8
2-2.5F		8	16	128	46	35	15	13½	1610	12.6
3-6F		18	18	324	42	42	12	12	1764	5.4
3-3F		12	12	144	36	36	12	12	1296	9.0
3-2F		10	10	100	34	34	12	12	1156	11.6
4-5F		11	22	242	46	35	12	12	1610	6.7
4-3F		9½	15½	147	39½	33½	12	12	1324	9.0
4-3F-4		9½	15½	147	23½	17½	4	4	411	2.8
4-3F-2		9½	15½	147	19½	13½	2	2	263	1.8
4-2F		8¾	11¼	98	35¼	32¼	12	12	1155	11.8
5-5F		8	31½	252	56	32	12¼	12	1792	7.1
5-3F		8	19	152	43	32	12	12	1376	9.1
5-3F-8		8	19	152	35	24	8	8	840	5.5
5-3F-4		8	19	152	27	16	4	4	432	2.8
5-3F-2		8	19	152	23	12	2	2	276	1.8
5-2F		8	12	96	36	32	12	12	1152	12.0
5-2F-8		8	12	96	28	24	8	8	672	7.0
5-2F-4		8	12	96	20	16	4	4	320	3.3
5-2F-2		8	12	96	16	12	2	2	192	2.0
5-2F-1		8	12	96	14	10	1	1	140	1.5
6-5F		11	22	242	46	35	12	12	1610	6.7
6-5F-8		11	22	242	38	27	8	8	1025	4.2
6-5F-4		11	22	242	30	19	4	4	570	2.4
6-5F-2		11	22	242	26	15	2	2	390	1.6
6-5F-1		11	22	242	24	13	1	1	312	1.3
7-5F	18			254	42	42	12	12	1764	6.9
7-3F	13¾			149	37¾	37¾	12	12	1425	9.6
7-2F	11¼			99	35¼	35¼	12	12	1243	12.5
8-6F		18	18	324	42	42	12	12	1764	5.4
8-3F		12	12	144	36	36	12	12	1296	9.0
8-3F-8		12	12	144	28	28	8	8	784	5.4
8-3F-4		12	12	144	20	20	4	4	400	2.8
8-3F-2		12	12	144	16	16	2	2	256	1.8
8-2F		10	10	100	34	34	12	12	1156	11.6
8-2F-8		10	10	100	26	26	8	8	676	6.8
8-2F-4		10	10	100	18	18	4	4	324	3.2
8-2F-2		10	10	100	14	14	2	2	196	2.0
8-2F-1		10	10	100	12	12	1	1	144	1.4
9-5F		11	22	242	46	35	12	12	1610	6.7
9-3F		9¼	15	139	39	33¼	12	12	1297	9.3
9-2F		8½	11¼	96	35¼	32½	12	12	1146	11.9
10-5F		8	31½	252	56	32	12¼	12	1792	7.1
10-3F		8	18	144	42	32	12	12	1344	9.3
10-2F		8	12	96	36	32	12	12	1152	12.0
11-5F	18			254	42	42	12	12	1764	6.9
11-5F-8	18			254	34	34	8	8	1156	4.5
11-5F-4	18			254	26	26	4	4	676	2.7
11-5F-2	18			254	22	22	2	2	484	1.9
11-5F-1	18			254	20	20	1	1	400	1.6
11-5F-1R	18			254	20*	20*	1*	1*	314	1.2
12-6F		18	18	324	42	42	12	12	1764	5.4
12-6F-8		18	18	324	34	34	8	8	1156	3.6
12-6F-4		18	18	324	26	26	4	4	676	2.1
12-6F-2		18	18	324	22	22	2	2	484	1.5
12-2F-2		10	10	100	14	14	2	2	196	2.0
12-2F-1		10	10	100	12	12	1	1	144	1.4
13-5F		11	22	242	46	35	12	12	1610	6.7
13-3F		9¼	15	139	39	33¼	12	12	1297	9.3
13-2F		8½	11¼	96	35¼	32½	12	12	1146	11.9
14-5F		8	31½	252	56	32	12	12	1792	7.1
14-5F-8		8	31½	252	47½	24	8	8	1140	4.5
14-5F-4		8	31½	252	39½	16	4	4	632	2.5
14-5F-2		8	31½	252	35½	12	2	2	426	1.7
14-5F-1		8	31½	252	33½	10	1	1	335	1.3
15-5F	18			254	42	42	12	12	1764	6.9
15-3F	13¾			149	37¾	37¾	12	12	1425	9.6
15-2F	11¼			99	35¼	35¼	12	12	1243	12.5
16-5F	18			254	42	42	12	12	1764	6.9
16-3F	13¾			149	37¾	37¾	12	12	1425	9.6
16-2F	11¼			99	35¼	35¼	12	12	1243	12.5
16-F-½R	8			50	9*	9*	½*	½*	64	1.3
17-5F	18			254	42	42	12	12	1764	6.9
17-3F	13¾			149	37¾	37¾	12	12	1425	9.6
17-2F	11¼			99	35¼	35¼	12	12	1243	12.5
18-F-10	8			50	28	28	10	10	784	15.7
18-F-8	8			50	24	24	8	8	576	11.5
18-F-4	8			50	16	16	4	4	256	5.1
18-F-2	8			50	12	12	2	2	144	2.9
18-F-1	8			50	10	10	1	1	100	2.0
18-F-1R	8			50	10*	10*	1*	1*	78	1.6
19-F	8			50	28	28	10	10	784	15.7
19-F-8	8			50	24	24	8	8	576	11.5
19-F-4	8			50	16	16	4	4	256	5.1
19-F-2	8			50	12	12	2	2	144	2.9
19-F-1	8			50	10	10	1	1	100	2.0

*Round flanges.

hoods having 8 in. diameter throats was 6 in. so that by overlapping the pipe and throat 1 in. the first tap was located one pipe diameter downstream from the throat and successive taps were located at 3 in. intervals up to 26 in. from the throat and the last tap at 41 in. from the throat.

The pressure drop across the orifice was measured by means of a vertical water manometer and the static suction readings were made with carefully calibrated inclined water manometers (see Fig. 1) in most instances. Vertical manometers were used from time to time as a double-check on the inclined gages. As a general rule, static readings on all hoods were made at one pipe diameter and three pipe diameters (actually 23 in.) from the throat at pipe velocities of about 2000, 3000 and 5000 fpm, and readings were made at all taps with a pipe velocity of 4000 fpm, a common duct velocity in industrial local exhaust systems. A large number of the readings at the 8 in. and 23 in. stations were made on both the single tap in the line of taps and on the interconnected taps spaced 90 deg apart. The purpose of this was to be certain that the single tap reading was correct, particularly with rectangular and square hoods where the static suction close to the throat is not the same at all points around the duct.

Hoods Studied

Data were collected for a variety of hoods most of which had 8 in. diameter throats facilitating convenient connection to the 8 in. duct diameter of the test equipment. A few hoods of different size were studied also. Suitable transformation sections and straight duct sections were used to permit attaching the different throat size hoods to the test equipment and to permit obtaining satisfactory air flow and suction readings. A 4.00 in. diameter orifice was substituted for the 6.00 in. one for the smaller hoods.

The hoods having 8 in. diameter throats were divided between round, square and rectangular; the rectangular group being further varied in regard to the ratio of the width to length of the hood face dimensions. Each shape group contained a variety of hoods as regards the rate of divergence of the taper, or the included angle. The ratio of the hood face area to the throat area was varied from about 5 (6½ for square hoods) to 2 (less for a few hoods) by cutting down the length of the hoods with shears.

The hoods were all made by the same sheet metal contractor. Twenty gage material was used and the *usual industrial construction* was specified. The throat sections were lap jointed and riveted. Not all the hoods were ordered at one time; they were made on at least four different orders at different times over a period of about three months. It is not likely that

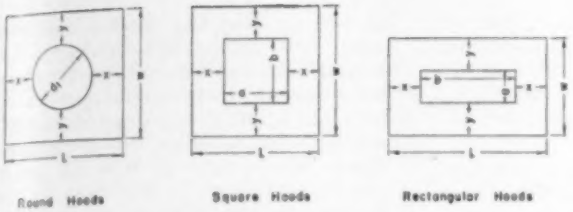


Fig. 10. Flanged hoods

they were all made by the same sheet metal worker since two distinct types of construction are apparent in Figs. 2 and 6. Hoods 16, 17, 20 and 21 present smoother taper to throat connection than do the others, yet it is the opinion of the authors that these hoods were representative of the type of construction usually encountered in industrial installations.

Seventeen of the hoods studied are shown in Fig. 6. This group shows the hoods in their original sizes, the face area to throat area being about 5 except for the square hoods which was about 6½. In Fig. 2 are shown the hoods after they had been reduced to the smallest face area to throat area ratio, about 2 for all except Nos. 1, 2 and 16 which were cut down even farther.. The intervening ratio hoods (4 and 3) are not shown. Flanging of the hoods was accomplished by attaching snugly fitting one piece sections of light weight wall board over the hoods so that they were flush with the edges of the hoods at the face.

Pertinent data on all the unflanged round hoods are given in Table 8. A total of 44 hoods of this type were studied resulting in comparable data for ten different rates of taper and for at least four different face area to throat area ratios on eight different hoods. As is shown in the table, the unflanged round group of hoods studied had tapers (included angles) of about 0, 10, 20, 30, 40, 50, 60, 90, 115 and 135 deg. A hood having a taper of 180 deg is really a flanged duct end and hoods of this type were included in the study (see Table 11). Greater tapers than 180 deg were not studied because they are seldom if ever used in industry. Furthermore, the coefficient of entry can be estimated sufficiently accurate for all practical purposes from the data for 180 and 0 deg (same as 360 deg) as indicated in Fig. 4. For a better understanding of the data given in Table 8, refer to Fig. 7. The hood numbers as used in Table 8 and other tables are significant in that the key or first number is the number assigned to the hood group as shown in Figs. 2 and 6 while the number following the dash is the approximate hood face area to throat area ratio. Thus 16-3 means hood No. 16 which has a face area to throat area ratio of about 3. Hood 16 shown in Fig. 6 is actually 16-5 while hood 16 shown in Fig. 2 is 16-1.2.

Four different unflanged, square hood groups and one individual hood were studied. Pertinent information on these hoods is given in Table 9 and Fig. 8. The included angles of this group were approximately 0, 40, 60, 90 and 115 deg. The face area to throat area ratios were the same as for round hoods except that for three hoods a ratio of about 6.5 also was studied. The numbering system for the hoods shown in Table 9 is the same as that for the hoods in Table 8.

Six different unflanged rectangular hood groups and two individual unflanged rectangular hoods were studied. Detailed information on these hoods is given in Table 10 and Fig. 9. Since in rectangular hoods the sides taper at two different rates, the included angles of both are given, the larger being the major and the smaller being the minor angle. The side ratios of the hood faces vary considerably even within the same hood group since the smaller hood sizes were made by cutting down the larger hoods. This procedure, while convenient from the viewpoint of cost, was not entirely desirable since it made difficult a good study of the effect, if any, of the sides ratio on the coefficient of entry.

It will be noted that in Tables 8, 9 and 10 most hood throat diameters (d_2) are given as 8 in. This is not strictly true since all 8 in. throat hoods excepting 16 and 17 had throats small enough to fit inside the 8 in. duct to which they were attached. Consequently, their effective inside diameters were slightly less than 8 in. They were, however, close enough to this dimension to permit ignoring it without introducing any significant error in the results. Hoods 16 and 17 on the other hand had 8 in. diameter throats and were, therefore, butted to the test duct and held in place with adhesive tape for study.

A total of 85 flanged hoods involving about 17 hood groups and two individual hoods were studied. Pertinent data on these hoods and flanges are given in Table 11 and Figs. 7, 8, 9 and 10. The numbering system is again significant in that the usual numbers for unflanged hoods are followed by the letter *F* for all flanged hoods and, unless otherwise indicated, the flange width was about 12 in. For other flange widths the actual width in inches follows the letter *F* and is separated therefrom by a dash. Most flanges on round hoods were square as indicated in Fig. 10, but a few of the narrow flanges on round hoods were circular following the outline of the hood edge. These are identified by the letter R following the flange width. Thus 11-5F-1R signifies hood No. 11 with a face area to throat area ratio of 5, a 1 in. wide flange which was round in shape.

Of interest to some readers are the data shown in Figs. 11 and 12 which give a fair idea of the air flow conditions obtaining in hoods of different rates of convergence and in similar hoods having different face areas.

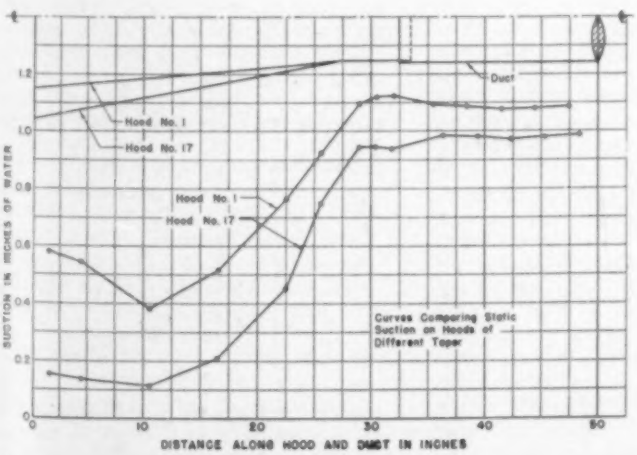

Duct
Hood No. 1
Hood No. 17
Hood No. 1
Hood No. 17
Curves Comparing Static Suction on Hoods of Different Taper
SUCTION IN INCHES OF WATER
DISTANCE ALONG HOOD AND DUCT IN INCHES

Fig. 11. Energy loss in hoods. Comparison of static suction on hoods of different taper

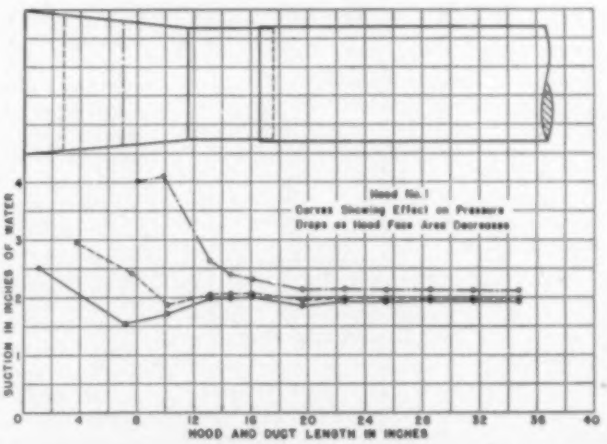

Hood No. 1
Curves Showing Effect on Pressure Drops as Hood Face Area Decreases
SUCTION IN INCHES OF WATER
HOOD AND DUCT LENGTH IN INCHES

Fig. 12. Energy loss in hoods. Effect of hood face area pressure drop

In Fig. 11 are shown the suction readings taken at various points on hoods 1 and 17, having included angles of 10 and 20 deg respectively and having face area to throat area ratios of 2.5 and 5, respectively. The curves are self-explanatory and show the influence of the vena contracta at the hood entrance, the static regain up to about 10 in. from the face, the influence of the increasing velocity as the air passes through smaller and smaller areas and finally the influence of the change in direction at the throat. The difference in the nature of the suction readings at the throat for the two hoods was caused by a difference in the throat diameters; hood No. 1 having a smaller diameter than the duct lying downstream thereof, whereas hood No. 17 had an effective inside diameter slightly larger than that of the downstream duct and was connected thereto by means of a butt joint as pointed out earlier.

In Fig. 12 are plotted a few suction readings taken on the same hood but in three different stages of face area reduction; face area to throat area ratios of 1.6, 1.4 and 1.2. Thus the top curve shows the readings when the hood was cut down to the size indicated by the *broken line and dash* line, the middle curve shows the readings when the hood was of the size shown by the *broken line* and the bottom curve shows the readings taken when the hood was the full size as shown in the figure. It is obvious of course, that the suction readings would not follow the straight lines joining the points where readings were taken. Data should have been collected at many more points to permit drawing accurate curves between them. The points of interest which are demonstrated by the data in the figure are the rate of increase in suction near the face as the area decreases and the following static regain so that the readings in the duct downstream of the hood are of the same order of magnitude. Had the face area to throat area all been above the critical value of about 2, the three curves would converge into one at or slightly before one pipe diameter downstream from the throat.

Discussion

W. C. L. HEMEON, Pittsburgh, Pa. (WRITTEN): A few years ago we prepared for the Industrial Hygiene Foundation a bulletin describing methods for measuring air flow in industrial exhaust systems, the most important portion of which dealt with the static suction technique.[11] We were inspired in this procedure by the excellent original exposition by John L. Alden.[12]

We desired to extend the data on entry coefficients beyond those given by Alden and concluded that coefficients could be derived by calculations based on hydrodynamic theory and shock loss data available in the literature. The coefficients that were presented were prepared by combining available data for the energy loss at the face of the hood with similar data applicable to the junction of the taper and the pipe, *i.e.*, throat.

Specifically, the coefficients resulted from an assumption that the loss at the face of an unflanged hood is 0.93 face velocity heads and that the loss at the throat is a function of the included angle. McElroy[13] cites values for the latter which we employed in part to derive our coefficients; the resulting coefficients agree well with experimental values reported by Dalla-Valle[14] and also with values from Gibson and Castel (quoted by Baumeister).

[11]See Bibliography, Note 5.
[12]See Bibliography, Note 1.
[13]See Bibliography, Note 7.
[14]See Bibliography, Note 4.

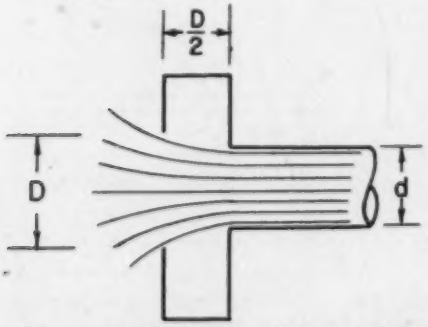


Fig. A—Theoretical suction opening designed for coincidence of throat with vena contracta $(D/d)^2 = 0.62$

The extensive test data obtained by the authors of this paper appear to be in disagreement with other data including ours. The following analysis of their data indicates that it is new data that need not be in agreement with others.

The authors, in their discussion, refer to included angle and to area ratios but have overlooked the taper length, a dimension of great importance. In cutting down the original hoods to vary the area ratio, they also changed the taper length and thus introduced two variables instead of one.

Nearly all the hoods studied by the authors are short. Only six have a taper length L exceeding one face diameter D. Our data were intended to be limited to hoods where the L/D is not less than 1 to 1½. Their data, therefore, extend the field of knowledge to shorter hoods, many of which are encountered in industry.

The importance of the L/D ratio is in its effect on the vena contracta just inside the hood face, the principle of which is illustrated in the special shape shown in Fig. A. The face and duct areas are related in the ratio of 1 to 0.62, which is the ratio of areas of a sharp edge orifice and its vena contracta. The distance between orifice and duct approximates the natural distance between orifice plate and the vena contracta. For such an opening we would expect a coefficient close to 0.98 or 0.99 (provided, further, that turbulence in the connecting chamber is insignificant).

In our view the short taper hoods described in the present study are identical in principle with the shape as suggested further in Fig. B. The contracted stream of air at the entrance, in both cases is directed into the duct and a major part of its energy is employed usefully, rather than being lost in shock inside the taper as would be expected in a taper long enough to complete the process of expansion.

In this analysis we have considered only the unflanged conical hoods, but the methods are equally applicable to the other shapes.

We have calculated the theoretical coefficient for hoods of different area ratios where no loss is assumed at entrance to the throat. Hoods with small included angle (*e.g.* 10 deg) closely approach this condition. The values, plotted in Fig. C, are obtained from the relation

$$C_c = \frac{1}{1 + 0.93\left(\frac{A_2}{A_1}\right)^2}$$

Also plotted thereon as circles are the authors' values for the coefficients of hoods with long tapers. They are in good agreement with our theoretical calculated values. The values of L/D range from 0.5 to 1.0. Values that appear as crosses and are well above predicted values are the subject of special attention in the following discussion.

At what point relative to the face of an unflanged hood is the vena contracta located, and what is the coefficient of contraction? These facts may be deduced from data given in the appendix showing static pressures at points along the taper. They indicate clearly that the vena contracta is located inside the face at a distance about 0.1 the face diameter and maintains itself up to about 1/3 of the face

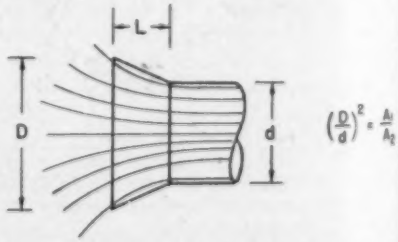


Fig. B—Hood with included angle of 45 deg where L/D is $0.4D$, or less, so vena contracta coincides with duct. Also A_1/A_2 is close to 1.85, the natural ratio of A_1/A_{vc}.

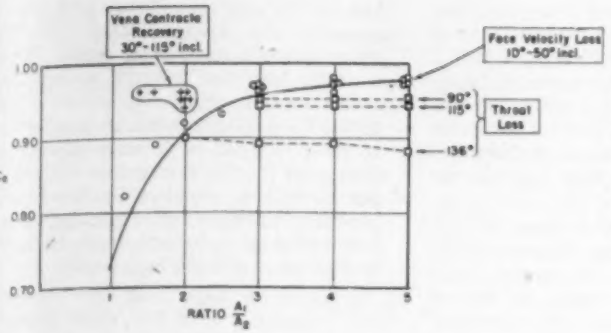


Fig. C—Coefficients arranged in 3 groups—those coinciding with theoretical curve — those where vena contracta velocity recovery is high — those where throat loss is controlling

diameter. It is also clear that recovery is complete at 0.6 to 0.8 *D*, average about 2/3 *D*.

A second fact of importance can be derived from these same data. From the static pressure of 0.58 in. just inside the face of hood 1, and related data, we can calculate that the coefficient of contraction is about 0.54, which may be compared with 0.62 for the sharp-edge orifice in a plate. Thus the area ratio A_1/A_{vena} is 1.85.

The significance of these factors may be summarized as follows:

1. If a hood has dimensions such that the length is more than about 0.6 *D*, enough space is provided so that expansion from the vena contracta effect is largely completed within the taper and, therefore, best agreement would be expected with theoretical calculations.

2. If the taper length is 1/10 *D* to 1/3 *D*, the throat will coincide with the position of the vena contracta, and its velocity energy will be utilized in the duct to result in a lower overall energy loss and a higher coefficient of entry.

3. If the A_1A_2 ratio of the hood coincides approximately with the ratio 1.85, that is the natural contraction of the entering stream, utilization of the velocity energy of the vena contracta would be greatest, provided that the conditions of (2) are also favorable, *i.e.*, a taper length of 1/10 to 1/3 *D*.

If we now examine the values that are unexpectedly high when compared with the theoretical, we see that all meet the conditions described in (2) and (3). They fall in the A_1/A_2 ratio of 1.5 to 2.0 and the distance from hood face to throat is from 0.1 to 0.4 (one 30 deg hood is at 0.5) the face diameter.

There remains for interpretation the values for the larger angles 90 deg, 115 deg, and 136 deg, at the ratios A_1/A_2 of 3, 4 and 5.

The energy loss resulting from contraction of the air stream at the throat of a hood becomes greater with increasing angles, and may become an important proportion of the total loss of large angles. The coefficients found for the large angles are fairly constant for all high ratios, at 0.95, 0.94 and 0.89 for 90 deg, 115 deg and 136 deg, respectively.

This independence of the coefficients relative to area ratios indicates that face velocity head loss is insignificant and that throat loss is, therefore, controlling. Coefficients of contraction of 0.76, 0.73 and 0.66 for 90 deg, 115 deg and 136 deg respectively, give entry coefficients that satisfy the experimental data.

In view of the foregoing considerations we would summarize our interpretation of the authors' data as follows:

1. Very few of the hoods tested are comparable with those for which previous data are published. For the most part they are shorter in taper length.

2. The data are readily correlated with theory by dividing them into three groups:

(*a*) Those in which the taper length is greater than 0.6 of the face diameter. (Includes angles only less than 60 deg.) For these the coefficient corresponds to a loss of 0.93 face velocity heads.

(*b*) Those in which the taper length is 0.1 to 0.4 the face diameter, and the ratio of face area to duct area A_1A_2 is between, approximately, 1½ to 2. For these the coefficient corresponds to a nearly complete recovery of face-vena-contracta velocity in the duct, giving values of 0.95 to 0.96. (Angles from 20 deg to 115 deg are represented in this group.)

(*c*) Those hoods with ratios A_1/A_2 of 3 to 5 and with included angles 90 deg through 136 deg. The coefficients for these hoods appear to be constant, *i.e.*, independent of A_1/A_2 and therefore the loss at the hood throat may be considered as controlling. Coefficients of contraction of 0.76 to 0.66 for these hoods satisfy the data adequately.

B. F. POSTMAN,[12] New York, N. Y. (WRITTEN): The data presented in this paper will be of practical value to designers of local exhaust systems. For years, designers have utilized their own experience plus a *factor of ignorance* in evaluating estimated losses of the various types of hoods utilized for industrial exhaust systems. Where adequate leeway had been allowed for assumptions or guess, relative to hood losses, the systems usually functioned in a manner which one would normally expect to result from the increase of static pressure allowed for necessary fan performance.

A discussion of the seven effects studied as outlined under *Conclusion* follows:

1. This item is important in that cramped quarters and involved piping are usually present when field tests are to be made of installed systems.

2. This item eliminates one *bogey* since the range of velocities covers the normal field of industrial exhaust application.

3. This item may be the cause for the ineffective results obtainable with many types of hoods installed to date. Structural considerations usually dictate the hood taper allowed for many installations. When these are not a factor, the hood appearance or the personal idiosyncrasies of the sheet metal man, unless constrained by specific dimensions, may be deciding factors determining the shape and size of the hood being installed.

4. This item is more or less a part of item 3.

5. It may be entirely possible that the importance of this item cannot be realized due to the small sized hoods tested. It does become a serious problem when large *fish-tail* hoods are installed on many involved industrial exhaust systems. Perhaps further research will be attempted to cover the larger hood installations.

6. There seems to be a practical limit to the application of flanges to small hoods, particularly portable hoods such as those used for the control of welding fumes. Flanges from 2 to 4 in. in width have been found acceptable by operators who handle small hoods as part of their daily operating procedure. As pointed out in the *Conclusion*, flanges provide an improved air flow pattern in front of the hood. In many instances, their use has cured a *sick* job.

7. There may be wider variations for this item than those indicated if the larger hoods are checked as mentioned under item No. 5 of this discussion. Some very interesting results might be forthcoming if the rule of thumb, 16 to 1, ratio for large hoods is exploded.

An interpretation of the results presented in this paper, possibly as a corollary to the average curves indicated in Fig. 4, but based on velocity head loss, would materially aid the designer to substitute *factors of research* for *factors of ignorance*.

As stated under "Purpose of Study" it is extremely important that the relationship between the exhaust ventilation rate and the contaminant concentration in the atmosphere be known for different hoods and different operations, if it is to serve as a guide on other similar operations. Research, such as this paper represents, tends to establish a practical approach to the most important phase of industrial exhaust as applied for the protection of the health of industrial workers.

LESLIE SILVERMAN,[13] Sc.D., Boston, Mass. (WRITTEN): I believe that Dr. Brandt and Mr. Steffy have presented data which will be of considerable value in the design and checking of industrial exhaust systems. It is interesting to note, from the data, the influence that turbulence at the inlet has on the coefficient of flanged duct ends. In Table 6, where the effect of flanged width is tabulated for different flange widths, it can be noted that the narrower flanges, particularly the round type, number 18F, create higher coefficients for duct ends. I believe

[12] Industrial Hygiene Engineer, Employers Mutual Liability Insurance Co. of Wisconsin.

[13] Assistant Professor of Industrial Hygiene, Harvard School of Public Health.

this may be explained by the diameter of the vortex created at the mouth of the inlet. The narrower the flange, apparently, the smaller is the radius of the vortex and hence lower inlet restriction and turbulence losses result. I am glad that the authors point out, however, that the entrance loss is not the only consideration in the selection of a hood, since the velocity characteristics in front of the opening are important. In the case of the narrower flange it can be predicted that it will not be as effective in velocity control in front of the opening.

It is of interest to note in Table 2 that the error made in air flow calculation with measurement at one pipe diameter is not serious. A comparison of the square roots of the two suction columns shows an error of 10.7 per cent for the unflanged round duct end, and 2.5 per cent for the flanged round duct end. These values represent the maximum error if only one pipe diameter is used and for field measurements, where length is limited, a correction factor could be applied if desired.

H. H. VALIQUET, Milwaukee, Wis. (WRITTEN): This research on losses through suction hoods has been comprehensive and practical in dealing with sheet metal fabrications reasonably representative of common practices. I saw many of these hoods on a recent visit to the Research Laboratory. My courteous reception by the staff as well as the frank discussion of the test results was appreciated.

There has been great need for this research as I pointed out to the Chicago Section of the *American Industrial Hygiene Association* in a paper entitled, Some Notes on Grinding Ventilation, etc., from which I quote: "The ratio of velocity pressure to total head in the pipe has been 0.5 and has not changed since published about 1910." This produces a coefficient of about 0.71 commonly used for grinding hood ventilation about which I was speaking. "It is not infrequently accepted, without just reason, for many dissimilar orifice arrangements. Tables of air flow are frequently shown in trade literature whether or not pertinent to the apparatus involved."

To ventilating engineers dealing with designs of reasonably correct controls for noxious and toxic substances this research is enlightening and very useful. An example of its value would be in fume hood exhaust design for welding hood No. 18F-2, where a coefficient 0.82 seems to apply to standardized flexible fume hood devices commercially available to which the manufacturer applies an 0.78 energy recovery.

In another field, that of banker exhaust for removing granite dust while finishing monuments, a coefficient of 0.72 is used which agrees with hood No. 18-1 coefficient 0.717. These research data will call attention to hood No. 1-1.2 the coefficient 0.823, of which should be attained in banker exhaust. One of the many facts brought forth in this excellent paper concerns hood No. 17-2 having a coefficient of 0.964, which provides an inexpensive fabricated sheet metal construction having a flow value equal to that of carefully machined ¾ thick orifices used on certain routine tests in high velocity exhaust orifices.

A very significant reference is made to the suction readings being reliable when taken one or more pipe diameters downstream from throat of hood. Where satisfactory performance is determined by use of suction readings, this distance is subject to local restrictions and some judgment.

Users of data presented in this paper will take into account many factors, such as unfavorable turbulences and mechanical obstructions, in adjusting their application of these fundamentals to special controls. There is, however, a large field for immediate application and adaptation of these data to uses in industry and they are invaluable where that delicate balance of control must not interfere with the mechanics of the process being performed.

W. N. WITHERIDGE, Detroit, Mich. (WRITTEN): It is a very great pleasure for me to comment on this history-making research report. As many of you know, I have maintained for some years that the Society's Research Laboratory has been overlooked far too long by engineers in the field of industrial hygiene. In Boston last year I expressed the opinion that industrial hygienists who did not understand the fundamentals of air flow were unwittingly using and perpetuating information that had never been subjected to a really well integrated study using sound methods by ventilating engineers who were competent to question the outworn data of earlier years.

Perhaps the best example of erroneous data that had enjoyed constant repetition in the literature is the implication that hoods designed with long tapers having included angles of 13 to 15 deg are the most efficient from the standpoint of energy loss. It is highly gratifying to find that the very common 45 deg taper, or 90 deg included angle, is at least as efficient as the 7 deg taper, or 14 deg included angle. As pointed out by the authors, this should be quite a relief to those sheet metal fabricators and hood designers who have rarely enjoyed the luxury of long entrance tapers.

It is high time also that the 0.71 entry coefficient be discarded. Fig. 4 in the report certainly indicates that the most likely range of coefficients for conventional industrial ventilating hoods is about 0.80 to 0.95. It also demonstrates that field inspectors who use static suction at the hood throat for estimating the rate of ventilation need more than ever to be advised that exhaust hoods producing the highest air flow rates are those with the lowest entrance losses and consequently the lowest static suction, other factors remaining equal. When a state law can specify a required static suction of five inches of water at the hood throat, without any regard to hood design, it is imperative that engineers fortify themselves with competent data as supplied by this report for the purpose of removing such uneconomical nonsense from the control of uninterested legislators.

I sincerely hope that this study is the beginning of a long and fruitful project in ASHVE Research on *industrial process ventilation*. It is not necessary of course that all such work be done at our own Laboratory, but it most certainly must be wisely coordinated by persons experienced in the field through the mechanism of the Society's Technical Advisory Committees and the Committee on Research. The Laboratory is to be commended for its tenacious pursuit of the general problem that was exposed by the view of many of our members at the special conference on this subject held in Detroit on March 13, 1945. We certainly have ample evidence from the survey conducted by the Special Advisory Committee on Industrial Ventilation that engineers around the country are hungry for sound and reliable design data, and will be ready to support any future work in this field with enthusiasm, and I hope, also with financial aid.

F. W. HUTCHINSON, Lafayette, Ind., stated that an overall conclusion to be reached from the paper would be that the hood design information used during the past 20 or 30 years was invalid. The paper disproved the basis of some of the design procedures which had been used.

J. N. LIVERMORE, Detroit, Mich., asked what type of pressure tips were used in obtaining static pressure readings in the tests.

A. C. STERN,[1] New York, N. Y., congratulated the authors on the excellent research job and the useful presentation of their results. He hoped that the work could be extended to cover hoods with wheels or tools in position so as to partially obstruct them in the manner most commonly found in practice.

He also presumed that the static pressure taps employed in this study were carefully drilled and finished smooth inside. In the field, he stated, it is common practice to test exhaust hoods by static pressure holes made by hammering the point of a punch through the duct wall, which results in internal denting of the duct and in ternal roughness.

Mr. Stern felt that it would be of interest to have the authors discuss the extent that air flow into a hood

[1]Chief Engr., Division of Industrial Hygiene, New York State Department of Labor.

will differ when the same apparent static pressure is read by manometer with each of these two kinds of static pressure taps.

AUTHOR'S CLOSURE: There appears to be little need for further comment either on the paper or on the excellent discussions except to thank the reviewers for their kind comments. From the comments of these reviewers, it is very apparent that there is great need for data of this kind by industrial hygiene engineers who interest themselves in controlling atmospheric contamination. This is only the beginning of what I hope may be a long series of similar studies which will provide the factual information needed to convert the engineering control of occupational diseases from an art to a science.

In particular, I want to thank Mr. Hemeon for his excellent theoretical analysis of the results, which, it is gratifying to note, checks the test results remarkably well. Certainly, his discussion is a most worthwhile contribution to the subject matter of the paper.

I regret, however, to point out that I am not in complete agreement with one or two of his statements. For example, we did not overlook taper length even though we did not analyze our data on that basis as he did so well. Taper length and face area or face diameter are so unalterably related that you cannot change one without changing the other unless you alter also the rate of taper. It is, therefore, impossible to have an L/d_1 ratio (see Fig. 7 of the paper) of more than 1 except for small included angles unless the face area to throat area ratio is increased well above 5. The relationship of L/d_1 ratio to the included angle for an area ratio of 5 is given in the following table:

L/d_1	Included Angle in Deg
1	31
1.5	21
2.0	16
2.5	13

Obviously, there is no point in increasing the area ratio beyond 5, and from the results in our paper it becomes questionable whether there is any practical value in studying hoods of such small taper rate that L/d_1 can be increased significantly above 1. In this connection, and in response to Mr. Postman's reference to the rule-of-thumb 16 to 1 ratio, it may be of interest to point out that we obtained data on one extreme hood which were not included in our paper. This hood had a 4 in. diameter throat, a 27 in. face, a 40 deg included angle, and a taper length of 31½ in. These dimensions make L/d_1 1.17 and the face area to throat area ratio 45. The static suction at one throat diameter differed by only two or three hundredths of an inch of water from that for the similar but much shorter hood listed in Table 7.

The other statement in Mr. Hemeon's discussion with which I cannot agree is item 1 in the summary. We specifically tested unflanged round hoods similar to 16 of the 19 given in the table in his paper[11]. Also our hoods were comparable to those referred to in our reference 4 in the Bibliography. What is more, they represent the type and size of hoods which are in very common use in industry.

[11]See Bibliography, Note 5.

"ADSORPTION"

This book[1] on a subject of great interest for ASHVE members was prepared as a correlation of the practical, commercial and engineering aspects of adsorption. It is intended to be of practical use to the designing engineer who may be concerned more with pressure drops, dry gas capacities and volume more than with theory, although the book contains a chapter on the Theories of Adsorption.

Many plant operators, equipment producers and material processors have cooperated with the author in providing data on processes and methods in practical commercial use.

The scope of the volume may be judged by the chapter titles, which include: The Unit Operation of Adsorption; Theories of Adsorption; Fuller's Earth and Activated Clays; Aluminum Oxide Base Materials; Bone Char and Related Materials; Decolorizing Carbons, Water-Treatment Carbon; Metal and Medicinal-Adsorbent Chars; Gas-Adsorbent Carbons; Silica Gel; Magnesia and Hydrous Oxides; Solvent Recovery and Adsorption from Gases; Odor Removal; Gas Masks; Gas Hydrates; Dehydration of Air and Gases; Ion Exchanges; Chromatographic Adsorption Analysis; Inspection, Specifications, and Testing of Adsorbents. Appendix — Fundamental Laws, Energy Relations, Conversion Factors.

[1]"Adsorption" by C. L. Mantell, Ph.D., Consulting Chemical Engineer, New York. Member of American Institute of Mining and Metallurgical Engineers, and American Institute of Chemical Engineers. Published by McGraw-Hill Book Co., Inc., New York and London, 1945.)

An Invitation to Members

The Council wishes to encourage authors to contribute good technical papers for presentation to our Society, particularly papers dealing with new or novel developments. Good papers on air conditioning of all kinds and new papers on air distribution, industrial hygiene and industrial ventilation are of interest. A special committee has been appointed to secure papers. The membership of the Committee on Technical Papers is composed of G. L. Tuve, chairman; L. T. Avery, Walter Heibel, C. H. Flink, and Cyril Tasker.

Any one of the committee members will be glad to consult by mail or otherwise with authors to predetermine whether the particular subject is desired.

Papers must be presented first to the Publication Committee for approval for publication and then are passed on to the Meetings Committee for selection to use at the Annual or Semi-Annual Meetings. We need papers planned now to take care of the Annual Meeting in January, 1947. If you have any papers you would like to suggest, please communicate with any of the committee members.

The Comfort Problem in Northern India

By W. C. Thoburn,* Ames, Ia.

SUMMARY—The author discusses the requirements for comfort in Northern India in summer and winter. The effects of massive wall and roof construction in absorbing diurnal variations in temperature are considered. Cooling of residence air by means of air tunnels located in the ground beneath houses is suggested for partial improvement of conditions. A test of an experimental cooling installation is reported.

PROBABLY most Americans are inclined to think of India as a hot, tropical country in which the comfort problem is almost entirely one of keeping cool. It may, therefore, be a little surprising to some to find on looking at the map that India, like Mexico, is divided roughly into two halves by the Tropic of Cancer, and that much of it is not really *tropical*. In fact, the northern plains, which are among the most populous regions, have a very definite heating problem in winter.

Climatic Conditions

India has a considerable variety of climates (see Fig. 1). In the far south the seasonal range of temperature change is small—roughly from 70 F on winter nights to 100 F on summer days—so that though cooling is desirable, the heating problem is almost absent. In the north (Fig. 2), and particularly the northwest, the extremes are much more pronounced and temperatures as low as 30 F and as high as 120 F are sometimes experienced. Though northern India winters are mild compared to the rigors of January with which most of us are familiar in the United States, the fact remains that, introduced to brick houses with little or no artificial heating, many an American newcomer has suffered more from cold during his first winter in India than ever before in his life. If his first reaction is to use his overcoat, he will probably very soon find it an inconvenience, for he will want to keep it on at his desk, since it will be very likely needed more if anything indoors than outdoors!

Another important climatic feature in India is the summer rainy season which comes with the southwest monsoon winds. Beginning rather abruptly, in June in the south and about July 1 in the north, the rains last for two or three months, radically changing the appearance of crops and vegetation generally, and covering large regions with a ceiling of clouds for weeks on end.

The rainy season brings lowered temperatures and of course increases the humidity considerably, and it is a question which is to be preferred, the hot dry weather of May and June or the cooler, but still quite warm, and moist weather of July, August and September. As a matter of fact, there are regional preferences which are officially recognized by provincial governments. Summer vacations for schools in Delhi and in the United Provinces to the east come just before the rains, and teachers who can afford it like to go to the Himalayas where the weather at that time is fortunately quite pleasant. West of Delhi in the Punjab, however, school vacations come during the rains, and those who go to the same hill stations during July and August will need to keep an umbrella always at hand.

The monsoon rainfall varies widely in different parts of India. In general it is heavy in the east, and where the winds meet rising mountain regions. The northwest is comparatively dry. Through most of India there is very little rain in winter, sunny days being the usual experience in cold weather.

Customary Ways of Promoting Thermal Comfort

Economic factors must, of course, be considered in the application of any residential comfort method. India is a country where some of the few wealthy are very wealthy, but where the great masses are extremely poor, and where the middle and professional classes would hardly compare in their standards of living with corresponding groups in the United States. Modern air conditioning for residences, including summer cooling, is still a luxury for the few even in America. Obviously, therefore, none but the wealthy in India can pay the costs of its installation and operation for their own homes. At present most people in northern India make their greatest adaptation to changes of weather by putting on

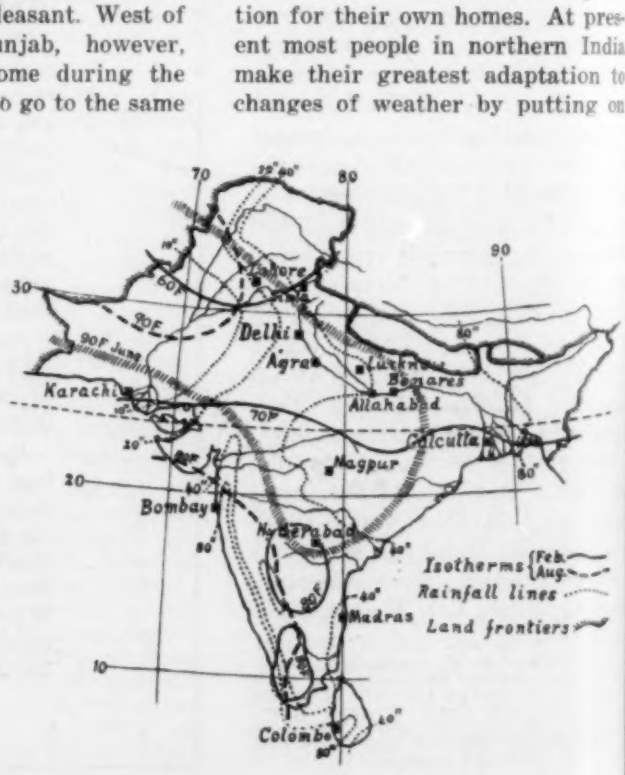


Fig. 1—Map of India showing climatic conditions

Data taken from the atlas of the Imperial Gazeteer of India. Courtesy of the British Information Services, New York.

*Graduate student at Iowa State College. Former Professor at Forman Christian College, Lahore, India.

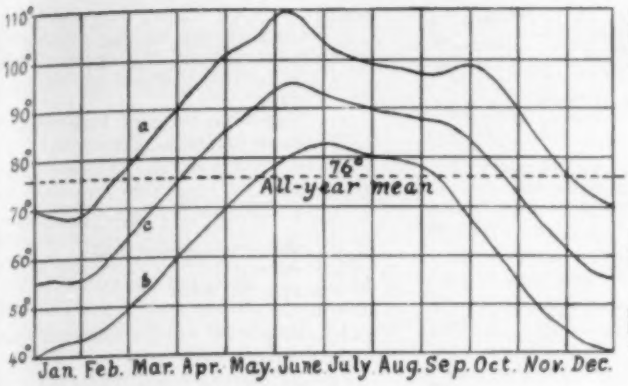


Fig. 2—Yearly temperature conditions in Lahore, Northwest India[1]

The graphs are based on the means of 63 years' observations (1878-1940)

a. 5-day mean maxima.
b. 5-day mean minima.
c. Means of a and b.

suitable clothing for the season concerned. The heat of summer and the cold of winter make them uncomfortable, and the efficiency of work in offices and stores is certainly not what it might be with proper air conditioning, but by and large, most people are not uncomfortable *enough* to spend very much of their personal income on means for improving their thermal comfort. The use of artificial methods of heating and cooling is therefore limited.

Comfort heating is usually obtained directly from fires. The fuel is generally the same as that used for cooking food, and includes coal, coke, charcoal, wood and cow-dung cakes, the last used mostly among the poorer and the rural groups. Fuels are burned in open braziers, fireplaces and occasionally in iron stoves. In many offices and some city homes electric radiators have become popular, but are not used by people of limited means. Gas as a fuel is almost confined to the Bunsen burners of laboratories and similar technical uses, and steam and hot water radiators are virtually unknown. The continuance of primitive methods of heating for comfort is due not only to the low economic level on which most people live and the lack of modern technology, but also to the fact that winter heating, like summer cooling, is not an absolute necessity. People *can* get along without it.

In summer those who can afford them use fans as an aid to air circulation. The fan of the past century consisted of a cloth fabric attached to a horizontal beam suspended from the ceiling of the room and pulled horizontally by means of a rope or cable (Fig. 3), coolie labor being employed. Incidentally, this type of work provided good jobs for blind men. Modern electric fans have very largely displaced the old swinging fan in the cities. Cooling by the evaporation of water is employed in a number of offices and larger establishments such as theaters. In its simple form this cooling is obtained by allowing the hot weather breeze to blow through a matting made of an indigenous fibre called *khass-khass* which is kept moistened by having water thrown on it frequently. In some places the wind and the wetting processes are mechanized, and in more and more of the larger buildings where the dryness of the climate warrants it, use is made of the more modern banks of water sprays operating in motor driven air streams.

Evaporative cooling has comparatively little value in the rainy season as previously mentioned, and during this period most business houses and other large institutions depend on ample ventilation and circulation of air to maintain reasonable comfort conditions.

Certain architectural features which are characteristic of the country and which are either directly related to the problem of comfort or are commonly believed to help in keeping houses cool in summer are found in the urban homes of northern India. Some of these features are:

1. Houses are usually built of masonry with thick walls and flat roofs. The bricks are held together with lime or cement plaster. Even earth-plaster without cement is often used to save costs in parts of buildings where strength is not important, and such *semi-strengthened* walls have often proved quite satisfactory. The roofs are most commonly laid on *jack-arch* sections of brickwork supported between parallel steel girders. A recent substitute for the jack-arch construction is the reinforced brick roof, where bricks and iron rods are laid in alternate parallel lines in the cement binding. In either case the roof is finished with an earth or clay layer and is usually topped with cement plaster or a layer of brick tiles.

2. Except in congested areas, most residences are of the single story type and tend to have spacious verandas which often extend more than halfway round the house. Verandas are frequently used for sleeping, but are also believed to be helpful in keeping the houses cool. It is a common summer practice to have beds moved out each evening to the veranda or even to the open yard. Some bungalows have cupola bedrooms built on the roof (Fig. 4).

3. Residential rooms are commonly built with high ceilings, the heights

[1]The Climate of Lahore, by P. R. KrishnaRao. (Punjab Geographical Review, 1942.)

Fig. 3—(Left) Woman coolie furnishing power for old swinging fan

Fig. 4—(Right) Typical bungalow in northern India

ranging from 12 to 22 ft above the floor. As these high ceilings are believed to be necessary for summer coolness, municipal regulations commonly specify the minimum heights of residential rooms. In Lahore, for example, it is stipulated that ground floor rooms shall not be less than 12 ft in height, and upstairs rooms not less than 10 ft.

4. Customarily there are many doors but few windows except the clerestory windows near the ceilings. These high-up windows are believed to be necessary for the removal of heated air. Many people practice the opening of doors and windows during summer nights, and closing them in the morning, in order to preserve the night coolness for the day. It is commonly thought that the night coolness is preserved by the entrapped air, and so the attempt is made to keep the day air out, except during the rainy season.

5. Frequently, though not invariably, the smaller *work* rooms of the house, such as the kitchen, pantry and bathroom, are provided with small windows and poor ventilation facilities. They are very rarely supplied with fans. These small rooms, therefore, have a reputation of being *stuffy* and *warm* in summer in comparison to the larger rooms, such as bedrooms, dining room and living room.

Architectural Factors Affecting Comfort

Any scientific approach to the problem of residential comfort should begin with a thorough study of the architectural factors that affect comfort. The manner in which summer and winter comfort may be enhanced by the way in which the house is constructed should be understood. When it was found that definite knowledge on the matter was very meager in India, experimental investigations were undertaken, together with some theoretical study, and in 1941 the following conclusions were reached as guiding principles for building for thermal comfort:

1. The most important requirement for summer comfort is liberal facilities for good ventilation. If dependence is placed mostly on natural ventilation, large windows are required in all rooms—windows which can be opened fully.

The value of ventilation in the Indian summer is amply supported both by scientific tests and everyday experience and practice. In a small closed room in July, the breathing and perspiration of the occupants were found to raise the relative humidity, and hence the effective temperature, rapidly to a very uncomfortable degree. Opening of windows and doors immediately brought relief. The same thing is illustrated by the marked difference experienced in crowded railway compartments in summer when the train is moving, as compared with the great discomfort felt when it is standing at the station. Except in the very few air conditioned cars, no one wants to travel with closed windows during this season in India.

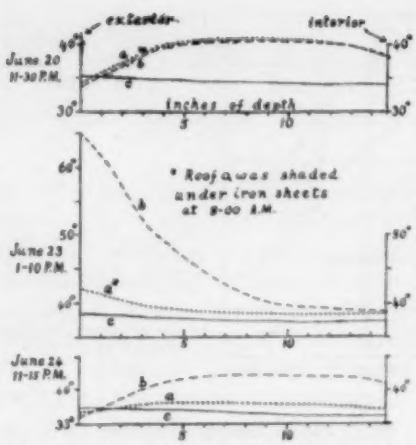


Fig. 5—Experimental observations of the temperatures resulting from heat penetration through three masonry roofs

In this and other experiments the c.g.s. system and Centigrade units were used.

It may be pointed out that, with effective temperatures in still air often well above 80 F, consciously felt air movement is usually very much welcomed. Most people can sleep soundly with a steady breeze from a table fan blowing directly over them all through a summer night.

2. Thick masonry effectively absorbs the diurnal variation of temperature, so that the inside of a 1 ft in thickness brick wall or roof tends to maintain a temperature close to the 24-hourly mean for its outside surface.

In June 1941 experimental tests were made to determine the varying temperatures within the masonry of three roofs. They were all between 14 and 15 in. in thickness, located in the same building and, except for one of them *c* which was protected by a second-story room above, they were similarly exposed to the sun. Temperatures were measured by means of thermocouples, and the results plotted. Typical temperature curves are shown in Fig. 5. Considering the fact that the *masonry* in the roofs was not homogeneous throughout the total thickness, and the irregularities of the weather, the results compare favorably with the equation for the penetration of a sinusoidal temperature wave into a solid from a plane boundary surface:

$$\theta = \theta_{max} e^{-\sqrt{\frac{\pi}{T}}\frac{x}{a}} \cos 2\pi\left(\frac{t}{T} - \frac{x}{2a\sqrt{\pi T}}\right) \quad \ldots\ldots (1)$$

where

θ = temperature at any depth
θ_{max} = maximum of the cycle of temperature change on the surface
e = Naperian base of logarithms
T = time period of the cycle

$$a = \text{penetrability} = \sqrt{\frac{k}{\rho s}}$$

k = thermal conductivity
ρ = density
s = specific heat
t = time
x = depth at which temperature is measured.

Curves plotted from Equation 1 are shown in Fig. 6a. In Fig. 6b are shown penetration curves based on the assumption that the solid, initially at a uniform temperature taken as zero has suddenly imposed on its plane surface a higher temperature θ_o which is maintained thereafter indefinitely.

These curves all show how thermal changes on the outside surface of a wall will be delayed in time and minimized in extent as the waves pass through to the inside surface. They also provide a means for computing the heat or cold stored in the solid by a given amount of warming or cooling of the surface. As indicated by the shading, the storage of heat or cold is found from the area under the curve. Estimating the storage of cold by overnight ventilation in summer on this basis, it may easily be shown that the benefits gained are due to the cooling off of the internal masonry surfaces in the house. The cold held by the entrapped night air is negligible in comparison.

3. Any method of shading roofs, walls, windows and doors from the direct rays of the sun will help to keep cool the regions so shaded. As long as the shading is effective, the material used for making the shade makes very little difference.

Evidence for this is shown strikingly in Fig. 5. The cooling effect of the sheet iron shade was almost as good as that of the heavy masonry above roof *c*. The iron sheets themselves became hot in the sun, but they were not red hot, and therefore radiated comparatively little from their under sides. And such secondary radiation would not pass through the glass of a window or door in appreciable amount.

4. Insulation is of value if artificial methods of heating or cooling are used. If no humanly controlled methods of heating or cooling are used, then the installation of insulating materials is virtually useless. This is a conclusion arising from the concept of heat transmission expressed in the formula for heat conduction. An insulating material discourages the transmission of heat by conduction. But if no means are used to maintain an inside temperature more comfort-

able than, and hence different from, the mean outside temperature, then there will be no net transmission of heat for the insulating material to discourage.

5. There is no evidence of any necessary relation between high ceilings and summer coolness for masonry residences. This conclusion is of a negative nature, and is the result of accumulated tests rather than direct proof. Of course there is no single physical reason that can be given for the dependence of the temperature of any object or enclosure on its dimensions. But it is necessary to deal with popular ideas which prevail. Consequently, a dozen conceivable theories for the supposed cooling effect of high ceilings were put down, commonly proposed explanations as well as others of varying degrees of plausibility. All these were examined carefully, and no effective cooling factor was found which was necessarily associated with high ceilings.

The common popular explanations are: (*a*) that the heated air rises to the top of the room in summer, (*b*) that the radiation from a low masonry ceiling in summer would be oppressive, and (*c*) that plenty of *air space* is needed (presumably for ventilation) in hot weather.

Against the first idea there is evidence of tests, showing that in summer weather there is very little thermal stratification of air in bungalow rooms. It is hardly to be expected, since there is virtually no source of *heated air* in the house in that season. Furthermore the use of the popular ceiling fan will eliminate any theory of rising hot air or heated air layers!

Against the second explanation both theory and tests will show that the added warmth due to direct radiation from a masonry ceiling is either nil or negligible. Verifying this is the common fact that *solar* pith hats, commonly worn for protection from the sun out of doors, are never wanted for head protection by even bald headed men inside masonry houses.

The third explanation is based on a confusion between air space measured in cubic feet, and the rate of ventilation measured in cubic feet per minute. Unfortunately it is not commonly appreciated that the rate of ventilation has no necessary relation to the volume of air in a room. The natural reaction to *stuffiness* calling for "more air" continues to be expressed architecturally by building large rooms with high ceilings. As might be easily expected, this conclusion has elicited some dispute, but though discussion has been freely invited, no objective evidence to the contrary has yet been brought forward. In 1943 a further practical experiment was carried out to test the possibility of a small room being comfortable in summer.

A store room in a bungalow was chosen, about 5 ft x 7 ft x 9 ft high. It had one small, screened window, and ordinarily would be considered quite unsuitable for hot weather sleeping. But it was equipped with an ordinary table fan, mounted on the window sill with a cardboard duct so arranged as to draw air directly from outside and blow it over the bed of the occupant. With this arrangement the little room was found at least as comfortable as any other room in the house for sleeping. It was used through June, the hottest month of 1943, by way of experiment. The next summer it was used again, this time by simple preference.

Of course it is not postulated that the list given is exhaustive or that there are no other constructional comfort principles applicable to Indian residences. On the other hand some of the conclusions which have been stated here may appear quite obvious. But sometimes it is necessary to state even the obvious. Mistakes in building frequently violate "perfectly obvious" principles. And a clear statement of the expected as well as the unexpected results is necessary in order to understand the whole investigation thoroughly.

Further Possibilities

The question now arises as to what further means of increasing residential comfort, or improvements in the present methods, can be employed in regions such as northern India where there is a comfortable mean between the extremes of seasonal temperatures. Any method to be widely used should be simple and fool-proof, and inexpensive to operate. Fortunately, the amount of heating and cooling called for is not great. A very welcome difference would be 10 deg F warming in winter and the same amount of cooling in summer, especially as most people manage to get along with much less. Complete air conditioning for residences is not needed for the present at least; and it may be well to emphasize that the best methods for meeting the comfort problem in India should be ultimately developed and studied in India, while working under Indian conditions.

As has been indicated, the present methods involving the evaporation of water could probably be considerably extended and improved in their efficiency. For home use there will presumably be a need for the mass production of effective but inexpensive units, operating on the ordinary electric house supply sys-

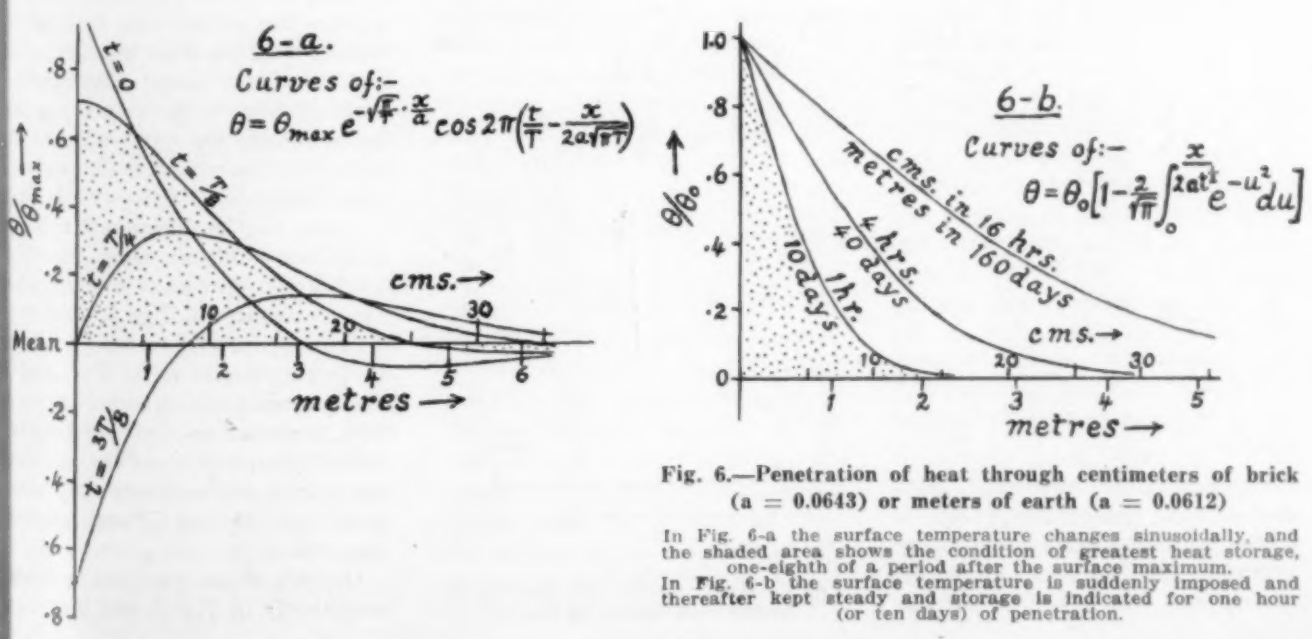


Fig. 6.—Penetration of heat through centimeters of brick (a = 0.0643) or meters of earth (a = 0.0612)

In Fig. 6-a the surface temperature changes sinusoidally, and the shaded area shows the condition of greatest heat storage, one-eighth of a period after the surface maximum.
In Fig. 6-b the surface temperature is suddenly imposed and thereafter kept steady and storage is indicated for one hour (or ten days) of penetration.

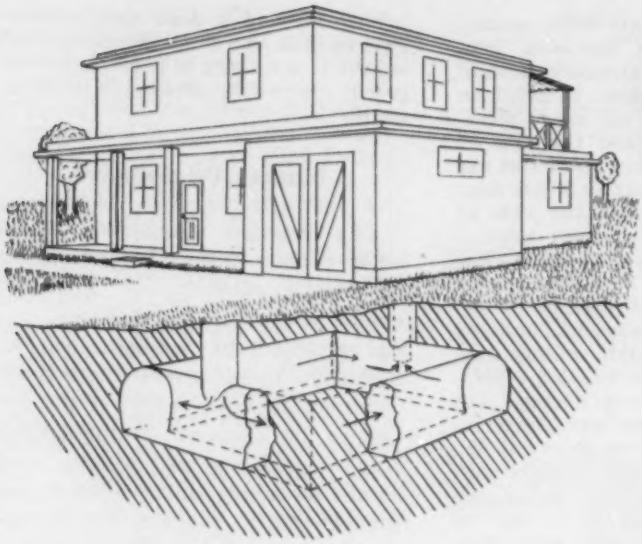


Fig. 7—(Left) Arrangement for cooling a house by means of underground air cooling tunnels

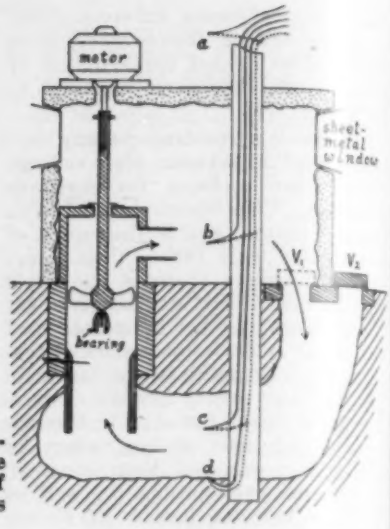


Fig. 8—(Right) Experimental house cooling by means of underground tunnels

tem. Still the fact remains that the benefits and effectiveness of such units would be limited to hot and comparatively dry weather.

For winter warming the possibility of solar home construction, using the direct rays of the sun, might well be explored. Here again the benefits would be limited, in this case to cold and at least nearly clear weather. Besides this, solar home construction would call for large glass windows which would displace considerable wall area on the south side of the house. It is possible that this might cause the loss of a considerable amount of the heat absorbing value of the masonry which tends, as we have seen, to maintain a mean temperature through the variations of day and night.

Another significant possibility is that of using earth stored heat and cold for enhancing residential comfort. Perhaps such a suggestion appears naive, and it is possible that it may not ultimately prove to be very practical. But it is a fact that underground temperatures are considerably cooler in summer and warmer in winter than air temperatures above the surface, as anyone who has inhabited a deep cellar will readily witness. And in the past underground rooms have actually been built and used in many places in India as aids to summer comfort.

By a combined process of conduction and absorption, heat is stored in the earth's crust every summer and drained therefrom during the winter. Most of this storage is in the upper layers, and at some 20 ft below the surface there is very little variation from the annual mean. This mean in northern India is a very comfortable temperature.

It is not proposed here that people live below ground level. The method suggested (Fig. 7) is to supply the occupant with an earth tempered air stream, not continuously but when and where he wants it, from a suitable system of insulated ducts. The air would come to the ducts from underground tunnels at an average depth of about 15 ft. In these tunnels, lined with masonry materials, the air would effectively acquire a temperature approaching that of the deep earth.

Estimates of the heat stored in the soil behind or under a given surface area may be made as indicated in Fig. 6, a and b. For a surface area A in the case of the sinusoidal cycle (Fig. 6, a) it is:

$$H = A\theta_{max}\sqrt{\frac{k\rho sT}{2\pi}}$$

In the case of the steady surface temperature (Fig. 6, b) it is:

$$H = A\,\theta_o\frac{2}{\sqrt{\pi}}\sqrt{k\rho st}$$

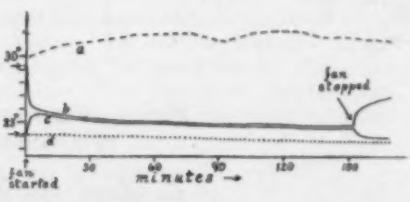


Fig. 9—Test results obtained with installation shown in Fig. 8

The amount of heat stored in the ground covered by a modest residence is considerable, and the heat or cold available from a tunnel surface area of about 200 sq yds is of the order of a billion calories or four million Btu. Of course, this supply would not be enough for the complete air conditioning of a house. But if it could be used for a little intermittent cooling during the three worst months of summer, and similarly for intermittent warming through the three worst months of winter, it might be a good deal better than the little now being done for residential comfort.

In order to try out the project experimentally, tests were carried out in 1944 with a miniature house. For convenience the diurnal cycle of temperature changes was substituted for the annual cycle. As will be seen from the formulas for heat penetration and storage, this reduction of the time scale by a factor of 365 called for a reduction in the scale of lengths by a factor equal to the square root of this figure, or about 19. The miniature house was accordingly built a foot high over a tunnel dug in the ground about a foot deep. The air could be circulated at will by a small motor-driven fan as shown in Fig. 8, and temperatures were measured by means of thermocouples at a, b, c and d which measured outside temperature, temperature of air striking occupant, temperature of air in tunnel and tunnel surface temperature respectively. V_1 and V_2 were controlling valves.

One set of observations is shown graphically in Fig. 9, and it is seen

that the results were in general very encouraging. The thermocouple junction b taking the place of the occupant was very successfully cooled by the circulating air during the heat of the day. However, before a full scale experiment can be expected to function successfully, certain problems should be solved, particularly in connection with the effectiveness of an earth tempered air stream projected upon an occupant. The project nevertheless indicates that whatever benefits may come from the use of earth stored heat and cold should be available at the very small operating cost of running a small electric fan. In the climate of northern India it should improve comfort both in summer and winter.

Scientific study of residential comfort problems under Indian conditions is rather in its infancy, and it is hoped that with the increasing popular interest in industrial technology and its development, this field of applied science will receive a much neglected share of attention in the years to come.

SZEKELY SELECTED BY *ESM*

Ernest Szekely, member of the Council of ASHVE, has been nominated for president of the Engineers' Society of Milwaukee for 1946-47, according to a recent announcement by E. L. Knebes, chairman of the *ESM* nominating committee.

Mr. Szekely, president of Bayley Blower Co., was elected *ESM* vice-president in 1945. He is also a member of *ASME* and past president of both Wisconsin and Ohio chapters of ASHVE.

Others nominated with Mr. Szekely are V. R. Tate for vice-president and for directors for a period of three years (1946-49) are L. D. Knapp, R. H. Earle and E. T. Sherwood. R. J. Panlener was nominated to fill the unexpired term of Mr. Tate as director.

1946 ANNUAL MEETING OF *ASTM*

The activity during the 1946 ASTM Annual Meeting held at the Statler Hotel, Buffalo, N. Y., the week beginning June 24 indicated the intensive interest in the Society's rapidly expanding work on specifications and tests for materials and methods of evaluating assemblies of these materials in various forms. The total registration for the week was 1,825, second highest for this annual event.

There were eight symposiums with technical papers ranging in number from four to fifteen covering a wide diversity of fields. The topics covered included: Symposiums on bearings, gas turbine materials, fatigue, spectroscopic light sources, oil procurement practices, testing parts and assemblies, pH measurements, atmospheric weathering of corrosion-resistant steels, and freezing-and-thawing tests of concrete.

In addition to these symposiums, there were other sessions devoted to effect of temperature on metals, cement and concrete, plastics, nonferrous metals, bituminous materials, and others.

One of the meeting features was the Edgar Marburg Lecture by Dr. J. J. Mattiello, technical director, Hilo Varnish Corp., on the topic *Protective Organic Coatings as Engineering Materials.* Dr. Mattiello's version of his complete lecture, which was shortened for presentation, was most interesting. The Nineteenth Award of the Charles B. Dudley Medal was made to H. R. Copson, research chemist, Research Laboratories, The International Nickel Co., Inc.

The high point of the meeting was the annual meeting dinner. Pres. J. R. Townsend gave his presidential address on *The Challenge of National and International Affairs to the Engineer,* and the guest speaker, Dr. B. K. Sandwell, editor of Canada's famous publication *Saturday Night,* spoke on *A Testing Time for Canada.* Both speakers stressed problems in connection with our democratic way of life, Mr. Townsend emphasizing the importance of engineers and technically trained people being public affairs conscious. Dr. Sandwell, who covered problems of our democracies from the national standpoint, emphasized the significance of the United States as a leader in developing changes in our present system which will insure continuance of the political philosophies which governed his country, the United States, Great Britain, and others.

There were 44 new tentatives approved during the meeting and a large number of revisions of existing standards and tentatives. The tentatives include proposed specifications and tests published for a year or more for comment and use prior to adoption as standard. Forty-two specifications and methods of tests were adopted as standard. These figures indicate tremendous activity in this field, not only in the newer committees such as plastics, etc., but in some of the older ones including steel, cast iron, and the like.

CHICAGO CONTRACTORS ELECT OFFICERS

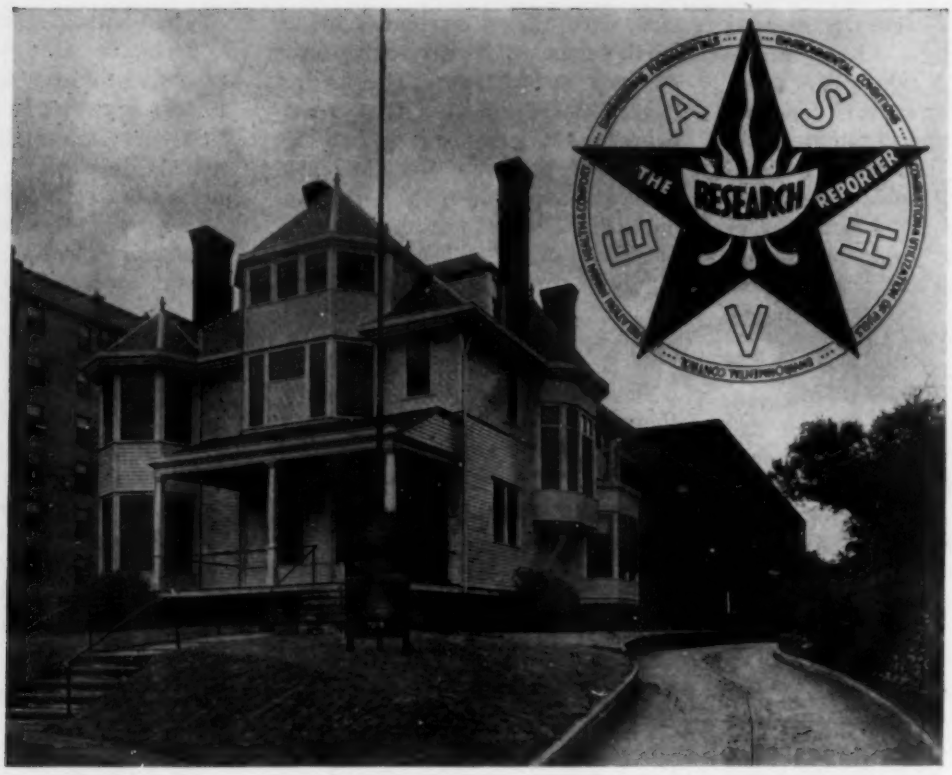
At a recent meeting of the *Heating, Piping and Air Conditioning Contractors' Chicago Association,* officers were elected as follows: Milton H. Westerberg, Northern Plumbing and Heating Co., president; C. H. Schlict, Westerlin & Campbell Co., vice president; C. W. Johnson, treasurer; Fred Dilzer, secretary, and L. T. Braun continues as executive secretary. The board of governors include R. W. Lanninger, W. F. Hildeman, J. A. Ljunggren and C. E. Crone.

Messrs. Braun, Crone and Johnson are ASHVE members and Mr. Westerberg starts his second term as president of the association.

NEW MANAGER FOR CHICAGO DISTRICT

George B. Coffey has been appointed manager of the Chicago district by A. M. Byers Co., with offices in the Conway Building, Chicago, and his operations will cover the states of Michigan, Minnesota, Wisconsin, North and South Dakota, Iowa, Indiana and parts of Illinois and northwestern Ohio.

He is a native of Freedom, Ind., and was graduated from the University of Indiana in 1924. He has been with A. M. Byers Co. since 1930. Mr. Coffey is president of the *Bridge and Building Men's Association* and a member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.



COMMITTEES MEET AT RESEARCH LABORATORY

With ample facilities for committee meetings and group conferences available in the new Laboratory at Cleveland, there has been considerable committee activity during July and August. Several meetings are planned for September.

AIR

The Technical Advisory Committee on *Air Distribution and Air Friction* held a full day meeting on July 29 under the chairmanship of Ernest Szekely. The Committee took positive action on the following:

a. The publication of a paper giving corrections to the present friction chart for temperature, viscosity and pipe roughness.

b. The modification of the present friction chart for round ducts to include smaller ducts and lower volumes.

c. An immediate start on a program covering friction in square and rectangular ducts and various types of fittings.

d. The publication of a paper on the throw of air from slots and jets based on previous research work at Case School of Applied Science.

e. The extension of the cooperative research program of air distribution under way at Case School to cover the effects of stack heads and other than isothermal conditions.

f. The approval of the cooperative research program at Kansas State College on the downward projection of heated air streams.

SOUND

The Technical Advisory Committee on *Sound Control* met at the Laboratory on July 30 under the chairmanship of R. D. Madison. A research project to be set up by the Material Laboratory, U. S. Naval Shipyard, New York, to determine the best methods for acoustical testing of ventilating fans and systems and to establish a method of predicting the acoustical performance of ventilating systems was submitted for Committee study and comment. The Committee on Research will be able to cooperate with the Navy Dept. in these tests through the Committee on Sound Control.

The Committee also recommended that the Society cooperate with other bodies in attempts to improve the present standards for the testing and calibration of sound measuring equipment.

Plans were made to initiate sound energy studies for unitary equipment as a cooperative research project.

RADIANT HEATING

A sub-committee of the Technical Advisory Committee on *Radiation and Comfort* met at the Laboratory on August 9 under the chairmanship of L. N. Hunter to plan a research program in radiant heating.

The Committee recommended that the Research Laboratory conduct a comprehensive library and field investigation to determine what work had been

done on radiant and panel heating, what was in progress and what was contemplated by the many interested organizations. The Committee strongly recommended that the effect of radiation on the comfort chart be investigated as quickly as possible.

A full meeting of the Technical Advisory Committee on Radiation and Comfort under the chairmanship of J. C. Fitts is planned for Saturday, September 21, at the Laboratory.

COOLING

The Technical Advisory Committee on *Cooling Load* met at the Laboratory on Friday, August 23, under the chairmanship of W. E. Zieber. Research programs on periodic heat flow and solar radiation transmission received careful study. A report of actions taken by the Committee will appear next month.

TECHNICAL ADVISORY COMMITTEES

THE Society's research program is established by the Committee on Research and supervised by the Director, under the authority of the Committee. To advise the Committee and the Director on specific projects Technical Advisory Committees are appointed by the Chairman. These Committees play a very important part in the research program. They are the medium by which members of the Society may suggest and comment on specific research investigations; they serve to correlate and coordinate industry and other opinions on research problems; they act as editorial boards for reports of Society research work and they approve all research papers prior to publication.

There are sixteen Technical Advisory Committees active in the present program. The title of the Committee, the name and address of the Chairman, and a brief summary of the scope of the Committee's activities are given. Members of the Society and others are invited to write to the individual Chairmen or to the Laboratory on matters related to the research program.

AIR CLEANING—R. S. Dill, National Bureau of Standards, Washington 25, D. C.: The characteristics of apparatus and equipment for the removal of atmospheric impurities. The development of methods for testing and rating air cleaning devices.

AIR CONDITIONING IN INDUSTRY—W. L. Fleisher, 475 Fifth Ave., New York 17, N. Y.: The role of atmospheric environmental conditions in industrial operations.

AIR DISTRIBUTION AND AIR FRICTION—Ernest Szekely, 1817 South 66th St., Milwaukee, Wis.: The factors involved in the distribution of heated and cooled air in enclosures. The resistance to air movement imposed by apparatus and equipment used in air distribution.

AIR STERILIZATION AND ODOR CONTROL—Prof. W. F. Wells, University of Pennsylvania Medical School, Philadelphia, Pa.: The study of the factors involved in air sanitation. The physiological and psychological effects of odors.

COOLING LOAD—W. E. Zieber, York Corp., York, Pa.: The study of all the components of the cooling load for enclosed spaces, especially periodic heat flow and solar radiation.

CORROSION—L. F. Collins, 14615 Prevost Ave., Detroit 27, Mich.: The mechanism and means of prevention of corrosion in all types of heating, ventilating, and air conditioning equipment.

FUELS—E. T. Selig, Jr., The Rybolt Heater Co., Ashland, Ohio: Factors in combustion and the efficient and economical utilization of all types of fuels used in heating appliances.

GLASS—R. A. Miller, 2200 Grant Building, Pittsburgh 19, Pa.: The study of winter heat loss and summer heat gain through glass in all its forms. The effects of shading.

HEATING LOAD—P. D. Close, 1736 Central Ave., Wilmette, Ill.: The characteristics of the components of the heating load for enclosed spaces.

INSULATION—Prof. E. R. Queer, Pennsylvania State College, State College, Pa.: The basic characteristics of insulating materials and their application.

PHYSIOLOGICAL RESEARCH—Dr. C.-E. A. Winslow, 313 St. Ronan St., New Haven 11, Conn.: The correlation of research on the physiological reactions of human beings to atmospheric environmental conditions. Its interpretation for the engineer.

RADIATION AND COMFORT—J. C. Fitts, 1250 Sixth Ave., New York 20, N. Y.: The study of the net exchange of radiant energy between an individual and his surroundings as it affects comfort. The development of heat transfer design data for use with the above.

SENSATIONS OF COMFORT—Prof. C. P. Yaglou, Harvard School of Public Health, 55 Shattuck St., Boston, Mass.: The factors which influence comfort and discomfort in heating and cooling.

SORBENTS—John Everetts, Jr., 212 Mission St., San Francisco 5, Calif.: The characteristics of liquid and solid materials used in the removal of water vapor.

SOUND CONTROL—R. D. Madison, 490 Broadway, Buffalo 5, N. Y.: The fundamentals of noise in occupied spaces resulting from air conditioning apparatus.

WEATHER DESIGN CONDITIONS—Thos. H. Urdahl, 726 Jackson Place, Washington, D. C.: The development of statistically sound weather data for use in the design of heating and air conditioning systems for enclosed spaces.

Hotel Statler

53rd Annual Meeting and Exposition in Cleveland in January

THE invitation of the Northern Ohio Chapter to hold the 53rd Annual Meeting of the Society in Cleveland, January 27-30, 1947, was accepted with pleasure by the Council, and members who attend will also have the opportunity of seeing the 7th International Heating and Ventilating Exposition at Lakeside Hall.

Announcement has been made by D. L. Taze, general chairman of the Committee on Arrangements that registration will be at the Hotel Statler. Assisting Mr. Taze as vice chairmen are E. B. Cary, Paul D. Gayman, F. A. Kitchen, and L. S. Ries.

Pres. Alfred J. Offner will preside at the technical sessions and the Committee on Arrangements plans for a series of entertainment functions to be climaxed by the annual banquet. Several inspection trips are being arranged and one of the most important points of interest for Society members will be the new Research Laboratory at 7218 Euclid Ave., where it is planned to hold various Technical Advisory Committee meetings.

There will be special events for the ladies and they will have the opportunity of visiting Cleveland's leading department stores, the Cleveland Museum of Art, which houses a notable collection of paintings, the Cleveland Museum of Natural History, Severance Hall, where concerts of the world famous Cleveland Symphony Orchestra are conducted, as well as downtown theaters and other points of interest.

It is easy to reach Cleveland by rail from all points in the United States via the Baltimore and Ohio, the Nickel Plate, Erie, New York Central, the Pennsylvania, and the Wheeling and Lake Erie R. R. Air lines connecting Cleveland with all parts of the country are American, Eastern, Pennsylvania Central and United. It is advisable to make early reservations for transportation.

All Society members have received special hotel reservation forms (blue) and these should be used as soon as possible in making hotel reservations. This form lists the various hotels and gives the room rates and should be mailed direct to the hotel chosen and confirmation will be made by the hotel.

EXPOSITION

The 7th International Heating and Ventilating Exposition will be the first since 1940 and it promises to be the best showing of heating and ventilating, air conditioning, refrigeration and allied products that has ever been assembled. Already 90 per cent of the available exhibit space has been taken and the leading firms in the field will show their latest equipment.

Exceptional interest surrounds the material to be viewed at the Exposition, as the past six years have produced an extraordinary number of new devices, many of which will be shown there for the first time. No less interest involves conventional equipment, a great deal of which has been redesigned to incorporate recent scientific advances or adjusted to meet the economic demands caused by altered manufacturing conditions. Such innovations affect not only individual machines, but in some instances involve entire layouts for installations in private dwellings and public

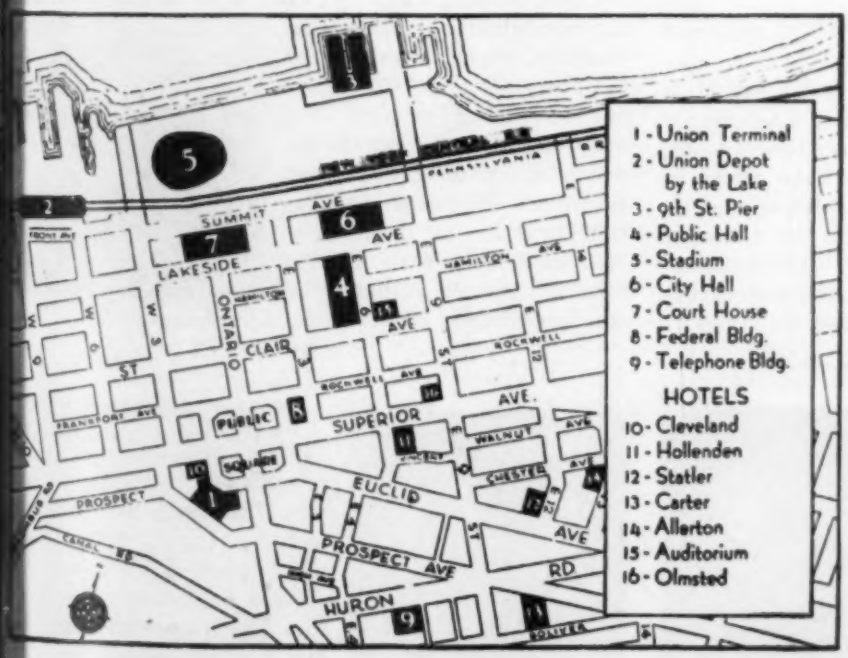
ablishments, while some of them will have a critical influence on the planning of industrial undertakings.

The exposition dates also correspond with those of the *National Warm Air Heating and Air Conditioning Association*, which will meet at the Hotel Cleveland, January 29 and 30, with committee meetings on the 27th and 28th.

Announcement has been made of appointments to the Advisory Committee of the Exposition, which is under the chairmanship of Alfred J. Offner, president of the ASHVE, and B. M. Woods and G. L. Tuve, first and second vice presidents of the Society, will serve as vice chairmen of the Advisory Committee, with the following ASHVE representatives: L. T. Avery, member of Council; L. P. Saunders, chairman, Committee on Research; H. R. Allenier, president, Central Ohio Chapter, Columbus, Ohio; Herman Seelbach, Jr., president, Western New York Chapter, Buffalo, N. Y.; J. E. Wilhelm, president, Northern Ohio Chapter, Cleveland, Ohio; A. W. Edwards, president, Cincinnati Chapter, Cincinnati, Ohio; F. R. Bishop, president, Michigan Chapter, Detroit, Mich., and John Paul Jones, consulting engineer, Cleveland.

Representing other organizations recognizing an interest in the Exposition are the following committee members: Stewart E. Lauer, president, *Air Conditioning and Refrigerating Machinery Association;* R. H. Hargrove, first vice president, *American Gas Association;* A. G. Syska, *American Institute of Consulting Engineers;* J. R. Edmunds, Jr., president, *American Institute of Architects;* C. S. Leopold, president, *American Society of Refrigerating Engineers;* Prof. G. Brooks Earnest, president, *Cleveland Technical Societies;* J. M. Frank, *Industrial Unit Heater Association;* J. B. Blackburn, managing director, *Insulation Board Institute;* Ernest Szekely, president, *National Association of Fan Manufacturers;* E. E. Dubry, representing *National District Heating Association;* E. I. Williams, president, *National Mineral Wool Association;* F. E. Mehrings, president, *National Warm Air Heating and Air Conditioning Association;* W. A. Matheson, president, *Oil-Heat Institute of America;* Marc A. Bluth, executive secretary, *Stoker Manufacturers Association;* W. R. Stockwell, *Institute of Boiler and Radiator Manufacturers,* and R. M. Getschow, *Heating, Piping and Air Conditioning Contractors National Association.*

General arrangements for the Exposition are being made by the International Exposition Co., New York, which has organized and conducted previous heating and ventilating expositions. Charles F. Roth is manager of the Exposition, and E. K. Stevens, associate manager.



Downtown Cleveland

TESTIMONIAL TO DR. WILLARD

At a *National Warm Air Heating and Air Conditioning Association* convention held in Chicago June 25 and 26, Dr. Arthur C. Willard, who retired as president of the University of Illinois on July 1, was presented a gold-framed testimonial by Frank E. Mehrings, president of the Association, in honor of 31 years devoted to warm air heating and air conditioning research.

A. C. Willard

In his speech following the presentation, Dr. Willard urged the association to continue its research program which began at the University of Illinois in 1917.

Dr. Willard is a past president and Life Member of ASHVE.

NEW OFFICERS ELECTED BY HOFFMAN SPECIALTY COMPANY

Announcement has been made of the retirement of Potter Bowles as president of the Hoffman Specialty Co. He will continue in the Hoffman organization as treasurer with headquarters in California. Under Mr. Bowles' direction the present plant and main office in Indianapolis were built and with this expansion program and a record volume of shipments in view, Mr. Bowles feels that he can move headquarters to California, where further development work will be carried out.

Following the July meeting of the board of directors, Paul G. Hoffman, chairman of the board, announced the election of Neil D. Skinner as president and the election of R. W. Cantin as secretary, who will carry out the policies and program of the company, founded by George D. Hoffman. Frank H. Gaylord continues as vice-president in charge of sales, and A. F. H. Scott remains as vice-president of the warm air division.

Who's Who in Navy Air Conditioning

IN THE building of a two-ocean Navy there were many engineering problems involving heating, ventilating, and cooling. More recently the preservation of the many units of the fleet brought special problems in the preservation of equipment which were solved by dehumidification methods. As this work was directed by the Bureau of Ships of the Navy Department, it is of interest to know something about the new thinking which produced extensive and permanent changes in the art of marine engineering, which involved heating and air conditioning. The personnel of the Bureau of Ships are proud of the results and it is significant that a good portion of the work was carried on by members of the ASHVE. At one time, nearly 75 per cent of the technical personnel in the air conditioning section of the Bureau were Society members.

The Society takes pride in this record of service and to indicate how extensive the contribution of its members has been in this phase of the Navy's work, a brief description of their activities is given.

DANIEL S. BERRES, (A 1943), head of testing and development group, air conditioning section. Mr. Berres maintains liaison between the air conditioning and contract design sections, and conducts performance studies on new ships, to assure close coordination of design concepts, shipbuilding, and operating practice. He specifies requirements and advises on techniques of ventilation performance testing.

LEON S. EPSTEIN, (J 1943), (former) head of dehumidification group, air conditioning section. Coordination of the activities of the many Naval and other organizations, who contributed to the development of dehumidification for the preservation of naval vessels, was carried on by Mr. Epstein until his recent resignation.

CLIFFORD C. CARSON, (M 1943), equipment development and design, air conditioning section. The design of each of the many pieces of new air conditioning equipment, for whose development the Navy was responsible during the past five years, first took shape under Mr. Carson's pencil.

ALBERT S. GATES, JR., head of dehumidification group, air conditioning section, (M 1943, A 1941). Mr. Gates, who handled ventilation of destroyers, submarines and several other types on the construction and repair group of the air conditioning section, recently took over the dehumidification group from Mr. Epstein.

A. EDWARD DESOMMA, (A 1943, J 1937), assistant head of demobilization section. Mr. DeSomma served in the construction and repair group of the air conditioning section, spent two years in New York as chief of ventilation section in the Office of Supervisor of Shipbuilding, and recently returned to the Bureau of Ships.

CLYDE M. HAMBLIN, (M 1943), chief technical advisor, air conditioning section. Mr. Hamblin, who has worked for the Navy for thirty-six years, in the period between the wars laid the foundations on which much of the present advanced shipboard ventilation practice is based. His rich experience now enables him to furnish guidance on special problems confronting any group of the section.

COMDR. JOHN EVERETTS, JR., (M 1938, A 1935, J 1929), west coast representative, air conditioning section. All the phases of the air conditioning section's responsibilities, including equipment distribution, and design and dehumidification consulting, have been carried on for the west coast by Comdr. Everetts.

RICHARD A. HEINDEL, (M 1943), head of equipment group, air conditioning section. The steady flow of equipment during the war from manufacturers to ships in service and under construction has been maintained by Mr. Heindel and his group.

COMDR. FERRY C. HOUGHTEN, (M 1921) (deceased) Bureau of Medicine and Surgery and Bureau of Ships liaison. Like Mr. Hamblin with ventilation, Commander Houghten was instrumental in bringing to fruition during the war his life work on the maintenance of adequate environment for working, in the form of the improvements now standard for Naval vessels.

LT. COMDR. H. R. IVERSON, (M 1936), equipment group, air conditioning section. Commander Iverson has administered the dissemination of design information on standard equipment to shipbuilding and repair activities. He has also taken part in much of the shipboard test work.

GEORGE P. LIVELY, (M 1942), head of ventilation and piping group, contract design section. The necessity for preparing the basic ventilation designs for new ships months or years in advance of building, places on Mr. Lively the responsibility to be abreast of new development in all phases of shipboard air conditioning, to insure that each new ship design takes the fullest advantage of progress and improvements.

JOHN G. MUIRHEID, (A 1940, J 1937), dehumidification group, air conditioning section. The scheduling of shipments so that each available piece of equipment made the greatest possible contribution to the ever-changing needs during the war and the demobilization has been handled by Mr. Muirheid.

CARL W. PORTER, (M 1943), construction and repair group, air conditioning section. Mr. Porter handles the ventilation of hospital ships, minecraft, PT boats and several other types. He also acts as a consultant on dehumidification for several of the inactive fleet bases.

COMDR. ELMER R. QUEER, (M 1933), administrative assistant, air conditioning section. In addition to his administrative duties, Commander Queer has taken a principal part in most of the shipboard test work and in the testing and development of cooling and dehumidification equipment.

LT. (JG) DAVID B. RODKIN, (A 1944), staff, 19th Fleet. Ensign Rodkin formerly handled the ventilation of a number of ship types in the Construction and Repair Group and is now handling dehumidification matters on the 19th Fleet Staff.

CAPTAIN ALFRED E. STACEY, JR., (M 1914), director, Material Laboratory, New York. Captain Stacey, who was assistant head of the air conditioning section early in the war, is now chief of the New York Material Laboratory, where much of the Navy's new equipment of all kinds is tested.

ROLAND J. THOMPSON, (A 1945), piping section. Mr. Thompson was formerly in the equipment group of the air conditioning section and handled matters pertaining to contracts, negotiations and purchases.

CAPT. THOMAS H. URDAHL, (M 1930), head, air conditioning section. Captain Urdahl has been in charge of the section since early in the war. He was largely responsible for the standardization which made it possible to design, provide and install the huge volume of ventilation demanded for shipbuilding, and initiated much of the testing and research which has resulted in basic improvements in equipment.

J. H. VAN ALSBURG, (M 1931), (former Lt. Comdr.).

Lt. Comdr. Van Alsburg was in charge of the equipment group when the wartime shipbuilding program was getting into high gear. He was then transferred to the staff of Commander Service Forces, South Pacific, and has since returned to private industry.

CON WHITTLESEY, (M 1941), (former head of testing and development group, air conditioning section). Mr. Whittlesey, who recently accepted a private position, had been in charge of the testing and development group since before the war, and played a leading role in most of the extensive shipboard tests conducted on new ship types.

COMDR. GEORGE C. WELLS, USN (applicant) prospective head of air conditioning section. Comdr. Wells, who has accomplished various assignments in the expansion of the Navy since the recommissioning of world war destroyers in 1939 and who was design superintendent at Pearl Harbor at the end of the war, will relieve Captain Urdahl upon the latter's resignation.

This program of providing effective heating and air conditioning for Navy vessels was carried out in both Navy yards and private yards. The work of this group contributed much to the efficiency of our Navy and many have returned to private industry where they are doing their part toward making this a better country in which to live.

In addition to the Society members who participated in these activities with the Bureau of Ships there were many others on the field staffs and also a long list of members who saw active sea duty.

PRESIDENT OFFNER GUEST AT ILLINOIS

The Illinois Chapter's final meeting of the season was held on May 13 at the Germania Club, preceded by a buffet supper, with Pres. Alfred J. Offner the guest of honor. President Offner spoke on the postwar era as a challenge to heating and air conditioning engineers.

The Chapter officers and committee chairmen summarized their activities during the 1945-46 season, and C. W. Johnson announced the results of the balloting for the coming year. The following officers were elected: President—O. J. Prentice; Vice-President—W. A. Kuechenberg; Treasurer—J. S. Kearney; Secretary—C. M. Burnam, Jr.; Board of Governors—C. E. Crone, P. J. Marschall, J. J. Philippi and J. S. Locke, retiring president.

Harry Short, Lafayette, Ind., gave a humorous yet instructive talk entitled "How Crazy Are You?"; his thesis was that everyone is *crazy* to some extent.

The Illinois Chapter ended a successful season showing a gain in membership. Every effort was made during the year to run the meetings on time, to present the various subjects in an interesting manner, and the routine chapter business was kept to the minimum to allow more time for the technical subjects and discussion.

SOCIETY OFFICERS TO VISIT CHAPTERS

Pres. Alfred J. Offner plans to visit Atlanta, October 7, when a meeting of the Council will be held and other stops in the south will be at New Orleans and San Antonio. He will continue on to the coast and address the Southern California Chapter, Los Angeles, October 14; Golden Gate Chapter, San Francisco, October 15; Oregon Chapter, Portland, October 17; Pacific Northwest Chapter, Seattle, October 18; and Manitoba Chapter, Winnipeg, October 21.

Dr. Baldwin M. Woods, first vice president of the Society, will attend the Atlanta Chapter meeting and en route will address the Kansas City Chapter, September 30; St. Louis Chapter, October 1; Delta Chapter at New Orleans, October 8; and South Texas Chapter in Houston, October 9.

Prof. George L. Tuve, second vice president, is scheduled to speak at the Iowa Chapter, Des Moines, November 4, Rocky Mountain Chapter meeting, Denver, November 6, and Nebraska Chapter, Omaha, November 7.

DEATH OF HENRY G. d'ISSERTELLE

Henry G. d'Issertelle, Tarrytown, N. Y., Life Member of ASHVE, former engineer with the General Electric Co., died at the Hillcrest Nursing Home, Ossining, N. Y., August 6, 1946, at the age of 78.

Mr. d'Issertelle was born in Charleston, S. C., and attended Massachusetts Institute of Technology. He retired fifteen years ago after thirty-five years as an electrical engineer with General Electric. While associated with General Electric he was a pioneer at the Sprague works of the company where much of the early experimental work in electrical engineering was performed.

In 1925 he won the company's Coffin Award for outstanding achievement among employees for work in his specialty field of heating and ventilating.

Mr. d'Issertelle, past commander of Knights Templar 47 in Yonkers, N. Y., and former member of the Common Council, lived in that vicinity for fifteen years and moved to Tarrytown five years ago. While living in Yonkers he was president of the Men's Club of St. Andrew's Church and belonged to the City Club, Fern Brook Lodge of Masons and Terrace City Chapter, R. A. M. He was a Shriner at Mecca Temple

in New York City and belonged to the Consistory there.

Mr. d'Issertelle, a close friend of the late Theodore Roosevelt, was a member of the Veteran Association, 1st Battalion, New York Naval Militia, which he joined in 1892.

Surviving are his daughter, Mrs. William E. van Wass, and two brothers, Norman A. and Charles D. d'Issertelle, to whom the Officers and Council of the Society extend their sincerest sympathy.

C. W. MILLER DIES IN MILWAUKEE

Word has just been received by the Society of the death on April 13 of Charles W. Miller, Milwaukee, Wis., president, The Rado Co.

Mr. Miller was born in Milwaukee in 1883 where he attended Milwaukee public schools and Marquette College. He also attended New York Trade School. His early experience was with the Lindemann & Hoverson Co. and the Thomas E. Hoye Heating Co., Milwaukee. With the latter he was assistant superintendent and draftsman, laying out and supervising the heating and ventilating work. He later organized his own firm for consulting and sales engineering.

Mr. Miller was one of the organizers of the Wisconsin Chapter of the Society and served as its first president in 1923. He also served as a member of its Board of Governors from 1924 to 1926.

It is with regret that the name of Mr. Miller is removed from the Society's roll and the Officers and Council extend their sincere sympathy to the surviving members of his family.

MARTIN H. HIRSCH DIES AT 43

The passing of Martin H. Hirsch, Forest Hills, N. Y., in May was a shock to his family and associates. Mr. Hirsch, engineer with the Consolidated Installations Corp., New York, N. Y., had been under the doctor's care for several months, but his sudden death was unexpected.

Mr. Hirsch, born in New York in 1903, was graduated from Mechanics Institute in 1922, from the College of the City of New York in 1923, and from Columbia University, where he specialized in air conditioning, in 1937.

He first became associated with Charles W. Barnett, plumbing and heating contractor, as estimator and superintendent, and in 1932 became a partner in the firm of Amann and Hirsch. He was also supervisor of the heating division of Sears, Roebuck & Co., White Plains, N. Y., before becoming sales engineer for Hoffman Specialty Co., Inc., New York. In this connection he designed and supervised installations of all types of heating equipment, including the Girls Dormitory, Mount Pleasant, Utah, and the housing development in Salina, Kans.

He was associated with J. P. Marsh Co. and The Trane Co., New York offices, and in 1943 he became assistant supervisor for Cox and Stevens, Naval architects in New York.

Mr. Hirsch is survived by his widow and two small children to whom the Officers and Council of the Society extend their heartfelt sympathy.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the names of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Committee on Admission and Advancement as soon as possible.

When the Committee on Admission and Advancement has acted favorably upon a Candidate's application and assigned his grade, the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 47 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Committee on Admission and Advancement, and in turn, the Council, urge members to assume their share of responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned.

All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by October 10, 1946, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary immediately after election.

ARNOLD, C. J., Engr., Hensol Castle Mental Deficiency Colony. PROPOSERS: R. H. Davies, W. A. Jones*. SECONDERS: Edward Lewis*, D. J. Thomas*.

AVILES, ALONSO, Mfgr's Agent. PROPOSERS: Leo Hungerford, R. J. Petersen. SECONDERS: R. A. Lowe, R. M. Storms.

BAUER, JOHN, Mgr., Bauer Heating & Plumbing Co. PROPOSERS: G. E. Merwin, D. E. McCulley. SECONDERS: B. G. Peterson, J. G. Bierl.

BILGREI, EMANUEL, Gen. Mgr., Marc Air Conditioning & Refrigeration Corp. PROPOSERS: H. E. Williams, George Belsky. SECONDERS: Seymour Brown, A. U. Zimmerman.

BLASE, THEODORE F., Draftsman, Air Conditioning & Engineering Co. PROPOSERS: C. A. Christierson, R. J. S. Caldwell. SECONDERS: Jacob Ehlers, S. H. Overton.

BRIDGMAN, RICHARD H., Gen. Mgr., Comfort Air, Inc. PROPOSERS: R. C. Cushing, C. L. Peterson. SECONDERS: D. I. Murphy, R. B. Holland.

CAPLAN, KNOWLTON J., Vent. Engr., Michigan Dept. of Health. PROPOSERS: K. E. Robinson, L. G. Miller. SECONDERS: V. H. Hill, A. D. Brandt.

CLINE, CARL C., Owner, Southern Refrigeration Co. PROPOSERS: W. H. Moler, J. A. Ray. SECONDERS: W. A. Catlett, H. J. Martyn.

DANNEHY, JAMES J., Engr., Refrigeration Maintenance Corp. PROPOSERS: M. J. Bamond, C. M. Burnam, Jr. SECONDERS: W. J. Baak, Fred Kaiser.

DOLLINGER, LEWIS L., JR., Vice-Pres., Dollinger Corp. PROPOSERS: F. J. Willson, J. S. Kilner. SECONDERS: J. W. McWilliams, R. E. Cherne.

DUNCAN, LOUIS P., Htg. Engr., George E. Trudel Co. PROPOSERS: J. F. Tuttle, A. L. Hare. SECONDERS: W. M. Russell, E. M. Palmer.

FAIRBAIRN, JOHN K., Mech. Engr., Kraft Foods Co. PROPOSERS: C. M. Burnam, Jr., J. S. Locke. SECONDERS: W. J. Baak, Fred Kaiser.

FELDMAN, MORTON M., JR., Mech. Engr., N. Y. Navy Yard. PROPOSERS: S. T. Hart*, D. E. Stearn*. SECONDERS: B. J. Lazan*, Max Ryen*.

*Non-Member

FRASER, G. M., Account Exec., Lawrence H. Selz Organization. PROPOSERS: C. M. Burnam, Jr., C. E. Price. SECONDERS: Thomas Cranage, J. W. Muessig.

GARDNER, ROBERT A., Sr. Engr., Norris Warming Co. Ltd. PROPOSERS: F. J. Whyte, W. Harding. SECONDERS: W. G. Chatterton White, R. R. Provan.

GOLLAHER, MONROE L., Sales Engr., Louis A. Roser Co. PROPOSERS: H. D. Miller, E. V. Gritton. SECONDERS: D. R. Wilde, H. G. Richardson.

HAHNE, W. H. E., Vice-Pres., Morf Industries, Inc. PROPOSERS: T. H. Urdahl, H. E. Parker. SECONDERS: J. W. Markert, E. E. Adams.

HAMAKER, H. J., Mgng. Dir., N. V. Technische Maatschappij Heringa & Wuthrich. PROPOSERS: M. P. deWilde, A. Silvera. SECONDERS: W. E. Vogel, R. S. Reynolds.

HELIN, WALTER E., Applica. Engr., Carrier Corp. PROPOSERS: J. H. Carpenter, O. W. Armspach. SECONDERS: R. H. Leveridge, Victor Cole.

HERKENHOFF, DONALD A., The David Ranken Jr. School of Mechanical Trades. PROPOSERS: J. T. Lynch, G. B. Rodenheiser. SECONDERS: B. C. Simons, H. C. Sharp.

HUDIK, FRANK, Pres., Comfortair Co. PROPOSERS: P. G. Greiss, O. C. Adams. SECONDERS: C. W. A. Steinmetz, E. R. Snavely.

KENT, E. ROBERT, Proj. Engr., Mehring & Hanson Co. PROPOSERS: T. A. Stokes, H. H. Hill. SECONDERS: W. H. Singleton, W. C. Jones.

KIDDER, KINSMAN B., Design Draftsman, The E-L-E Co. PROPOSERS: Leo Hungerford, E. C. Evans. SECONDERS: C. S. Anderson, F. L. McDonald.

KLINE, W. O., Air Cond. Div., Frick Co. PROPOSERS: E. H. Taze, P. J. Vincent. SECONDERS: A. W. Nugent, E. L. Crosby.

KYTE, ROBERT L., Chief Engr., Williams Heating Engineers. PROPOSERS: H. F. Eastwood, D. J. Calnan. SECONDERS: S. A. S. Patorno, J. A. Darts.

LA POINT, KENNETH K., Mgr., Charles M. Bailey Co. PROPOSERS: H. V. Hickman, G. M. Simonson. SECONDERS: P. R. Babcock, W. W. Cockins.

LEVINSON, DAVID S., Grad. Student, University of Illinois. PROPOSERS: S. Konzo, A. P. Kratz. SECONDERS: J. R. Fellows, R. W. Roose.

LINDENAUER, MARTIN M., Jr. Engr., Mance Corp. PROPOSERS: B. H. Lee, Francis Ficarra. SECONDERS: J. C. Mance*, F. J. Walter*.

LOW, CHARLES J., Engr., Jackson & Church Co. PROPOSERS: L. G. Miller, E. B. Root. SECONDERS: C. H. Pesterfield, J. H. Van Alsburg.

MALM, EDWIN L., Sales Pro. Mgr., Bell & Gossett Co. PROPOSERS: E. N. McDonnell, W. T. Miller. SECONDERS: E. J. Gossett, L. H. Harrington. *(Advancement)*

MCCAULEY, DALE G., Pres., Lansing Heating & Ventilating Co. PROPOSERS: R. A. Parsons, L. A. Tilford. SECONDERS: L. G. Miller, C. H. Pesterfield.

MORETTI, GARNI, Htg. & Air Cond. Engr., Lewis Engineering Service. PROPOSERS: Daniel Lewis, R. A. Oberschulte. SECONDERS: R. D. Randall, A. E. Knibb.

MOREY, ALBERT A., Asst. Vice-Pres., Marsh & McLennan. PROPOSERS: Emil Haas, Jr., H. E. Gould. SECONDERS: A. J. Natkin, C. D. Bridges.

OKRAY, RALPH A., Engr., Kleenaire Corp. PROPOSERS: J. E. Seymour, W. H. Feirn. SECONDERS: George Hall, Rex Vernon.

PALMER, NEWTON A., Engr., Eureka-Williams Corp. PROPOSERS: O. E. Nesmith, C. E. Bronson. SECONDERS: J. M. Hartman, E. M. Palmer.

PHILLIPS, RALEIGH C., JR., Student, The David Ranken Jr. School of Mechanical Trades. PROPOSERS: J. T. Lynch, G. B. Rohenheiser. SECONDERS: B. C. Simons, H. C. Sharp.

REBMANN, CHARLES P., Drafting Engr., John D. Falvey. PROPOSERS: J. D. Falvey, H. C. Sharp. SECONDERS: E. P. Harder, W. G. Frochey.

SCHERRER, JAMES R., JR., Ch. Engr., Security Manufacturing Co. PROPOSERS: P. C. Leffel, W. A. Reichow. SECONDERS: S. L. Furber, Henry Gould.

SHANAHAN, JOHN J., Contractor. PROPOSERS: C. A. Neff*, T. D. Fitzgibbon*. SECONDERS: H. D. Gilson*, C. J. Lindeman*.

SHEPPERD, W. B., Chief Engr., Koolaire Inc. PROPOSERS: E. H Colestock*, W. F. Hauber*. SECONDERS: L. A. Griffith*, O. L. Lilja.

TYE, JAMES L., Asst. Engr., C. Wallace Plumbing Co. PROPOSERS: L. S. Gilbert, T. H. Anspacher. SECONDERS: R. G. Lyford, B. S. Foss, Jr.

URAN, NEJDET F., Engr., T.B.M.M. PROPOSERS: Axel Marin, H. E. Keeler*. SECONDERS: B. F. Raber, F. W. Hutchinson.

VARMING, JORGEN, Cons. Engr., Steensen & Varming. PROPOSERS: W. E. Reck*, Mogens Balslev*. SECONDERS: N. C. Geersten*, Niels Steensen*.

WALKER, WILLIAM S., Dvlpt. Engr., H. H. Robertson Co. PROPOSERS: E. A. Saibel*, J. M. Rice*. SECONDERS: H. H. Crafton*, A. W. Coffman*.

WEGER, LEONARD, Cons. Engr., Cronheim & Weger. PROPOSERS: F. H. Buzzard, Nathan Cronheim*. SECONDERS: C. S. Leopold, F. R. Kelble.

WOLFE, BENJAMIN M., Air Cond. Engr., General Air Conditioning Corp. PROPOSERS: Leo Hungerford, Carlyle Artran. SECONDERS: J. B. Venneman, R. A. Lowe.

YOUNG, DICK H., Partner, Young Heating Co. PROPOSERS: W. W. Stuart, B. E. Landes. SECONDERS: W. H. McGuiness, D. C. Murphy.

*Non-Member.

CANDIDATES ELECTED

In the past issues of the Journal of the Society the names of the following men were listed as Candidates for Membership. The membership grade of each Candidate has been assigned by the Committee on Admission and Advancement and balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. B-III, Sec. 8, of the By-Laws, the following list of candidates elected:

Members

BARTELS, EVERETT M., Supvsr., Independent School District, Des Moines, Ia. *(Advancement)*.

BREY, JOHN O., Cons. Engr., de Luca & Assocs., Jacksonville, Fla.

CHASE, ROGER E., Sr., Pres. & Mgr., R. E. Chase & Co., Inc., Tacoma, Wash. *(Advancement)*.

CLAPPERTON, ROBERT, Proj. Engr., Dominion Rubber Co., Ltd., Montreal, Que., Canada *(Advancement)*.

COLLINS, JOHN W., Chief Engr., National Heating Products, Ltd., Montreal, Que., Canada *(Advancement)*.

CURRY, WILLIAM H., Chief Engr., Milledgeville State Hospital, Hardwick, Ga.

DERBY, MALCOLM R., Engr., Kroeschell Engineering Co., Chicago, Ill.

DOUDICAN, CHARLES T., Htg. Engr., M. J. Gibbons Supply Co., Dayton, Ohio *(Advancement)*.

DUVAL, ELAIR D., Htg.-Vent. Engr., A. L. Wright Jr. & Co., Portsmouth, Va.

FURBER, STANLEY L., Sales Engr., Minneapolis-Honeywell Regulator Co., Kansas City, Mo. *(Advancement)*.

GREEN, EDGAR L., Mech. Engr., Eastman Kodak Co., Rochester, N. Y.

HAMLIN, MILLARD R., Dist. Repr., Warren Webster Co., Columbus, Ohio.

HEIMAN, SIDNEY J., Pres., International Oil Burner Co., St. Louis, Mo.

HIGHTOWER, GEORGE B., Dist. Mgr., General Electric Co., Atlanta, Ga.

HOFFMAN, HERMANN C., Engr., Carrier Corp., Chicago, Ill.

IBISON, JAMES L., Appl. Engr., Ralph B. Johnson & Co., Houston, Tex. *(Advancement).*

JOHNSON, BLAINE A., Exec. Engr., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

KLEIN, WALTER A., Kremer-Hicks Co., St. Louis, Mo. *(Reinstatement).*

LABUS, OTTO A., Lab. Dir., The Trane Co., La Crosse, Wis.
LUCE, JOHN A., Mgr., Htg. Dept., Federal Pipe & Supply Co., San Francisco, Cal.

MARTENIS, WILLIAM W., Chief Co-ordinating Engr., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.
MCCANDLESS, HOWARD F., Mgr., Juillard Frozen Food Corp., Watsonville, Cal. *(Advancement).*
MENENDEZ, RAMON A., Prof. Engr., Tampa, Fla.
MILLER, EDWIN B., Sales Engr., C. A. Dunham Co., Birmingham, Ala.

RESS, OTTO J., Research Dir., L. J. Mueller Furnace Co., Milwaukee, Wis. *(Advancement).*
RONALD, EDWARD R., Partner, Warren & Ronald, Cons., Engrs., Louisville, Ky.

SUNDEEN, EARL E., Supvsr. & Designer, International Harvester Co., Chicago, Ill.
SWANEY, FRANK J., Asst. Vice-Pres., Cargocaire Eng. Corp., New York, N. Y.

TERHUNE, RALPH D., Mgr., Bryant Heater Co., Ridgewood, N. J. *(Advancement).*
THOMPSON, CLIFFORD J., Pres., Thompson Research & Mfg. Co., Davenport, Fla.

WELSHMAN, HUGH, Jr., Mgr., Grinnell Co., Inc., Providence, R. I.
WHEELER, JAMES A., Branch Mgr., American Blower Corp., Houston, Tex.

Associate Members

ANNABEL, CECIL H., Owner, Belsmith Co., Las Vegas, Nev.
AYRES, ROBERT O., Owner, Ayres & Co., Houston, Tex.

BAYLESS, ROBERT L., Jr., Owner, Valley Sheet Metal Co., Phoenix, Ariz.
BEDARD, EARL L., Proj. Co-ordinating Engr., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.
BELL, D. G., Gen. Mgr., North Texas Air Conditioning Co., Dallas, Tex.
BOYENTON, JOHN F., Pres., Refrigeration Suppliers, Inc., Norfolk, Va.
BOYLSON, JOHN, Supt., Valley Sheet Metal Co., Phoenix, Ariz.
BURTON, WALTER A., Sales Engr., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.

CHIDESTER, CHARLES W., Mgr., Pacific Scientific Co., Los Angeles, Cal.
COLEMAN, JAMES L., Jr., Vice-Pres. & Secy., Sauer, Inc., Pittsburgh, Pa.
COMBS, CHARLES F., Salesman, Russel J. Smith, St. Louis, Mo.
CROCKWELL, WALTER H., Salesman, Service Heat & Power Co., Oakland, Cal.

DE WILDE, M. P., The Hague, Holland *(Reinstatement).*
DICKEY, ALBERT T., Supvsr., Cox & Blackburn, Houston, Tex.
DOEPPENSCHMIDT, LAWRENCE C., Owner, Better Htg. & Air Cond. Co., Houston, Tex.

FEREDAY, MAURICE, Owner, Maurice Fereday Plbg. & Htg., Pocatello, Idaho.
FOXHALL, WILLIAM B., Asst. Ed., Heating & Ventilating, New York, N. Y.
FREDERICK, VERNON L., Capt., A. P. Hill Military Reservation, Fredericksburg, Va.
FREEMAN, LEWIS E., Mgr. Htg. Dept., Sullivan Hardware Co., Greenville, S. C.
FULLERTON, LAURENCE I., Pres., The Colton Corp., Kankakee, Ill.

GERRISH, DAVID C., Appl. Engr., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.
GRAVES, ROBIN A., Owner, Graves Engineering Co., Lynwood, Conn.
GRIM, FREDERICK, Service Sales Engr., Hajoca Corp., Roanoke, Va.
GRUBBS, JOHN K., Mech. Engr., Kroeschell Engineering Co., Chicago, Ill.

HARRISON, HOWARD W., Htg. & Ind. Power Engr., Georgia Power Co., Atlanta, Ga.
HARRISON, MAURICE E., Htg. Engr., A. Brizard, Inc., Arcata, Cal.
HECKLER, FRANK W., Heckler Bros., Pittsburgh, Pa.
HOLLADAY, WILLIAM L., Chief Engr., Airtopia Distributors, Inc., Los Angeles, Cal.
HORN, EDWARD C., Engr., J. M. Gallagher Co., Nashville, Tenn.

JENKINS, HARRY C., Sales Mgr., Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.
JONES, DAVID C., Mgr., Service Heat & Power Co., Oakland, Cal.
JUNG, FRANCIS P., Hygiene Engr., Tennessee Dept. of Public Health, Nashville, Tenn.
JUNGER, HENRY G., Sales Engr., Anemostat Corp. of America, New York, N. Y.

MACLEAN, IAN R. S., Engr., Carrier Lix Klett, Buenos Aires, Argentina.
MASON, LAWRENCE T., Asst. Mech. Engr., Board of Education, Detroit, Mich.
MCCALMON, ROBERT T., Field Service Engr., A. M. Byers Co., Pittsburgh, Pa.
MINDA, WALTER F., Sales Engr., Fla. Hill York Corp. Miami, Fla.
MOSHER, WILLIAM O., Asst. Mgr., Sumner L. Willson, Hartford, Conn.
MULLEN, C. HERBERT, Architect, Joplin, Mo.
MUNRO, W. J., Pres., Munro Motor Co., Inc., Rolla, N. D.

NASSIR, NICHOLAS, Design Engr., C. F. Braun & Co., Alhambra, Cal.

PEARSON, VERN E., Supvsr., Minneapolis Gas Light Co., Minneapolis, Minn.
PHILLIPS, D'ARCY E., Design Engr., Newport News Shipbuilding & Dry Dock Co., Newport News, Va.

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TOM BROWN

The many friends of Tom Brown will learn with regret of his passing at the age of 51 on Monday, August 26th, after a long illness from which everyone had hoped for his complete recovery.

He was born in Chicago on July 27, 1895 and graduated from the New Trier High School, and then entered the University of Illinois where he completed three and a half years of an architectural engineering course before World War I interrupted his studies. He served with distinction in the Infantry overseas and was discharged as a captain at the end of the war.

His first business experience was with Graham, Anderson, Probst and White, where he worked on the layout of heating and ventilating designs for the Field Museum and Union Station in Chicago.

Tom Brown

Mr. Brown joined the B. F. Reynolds and Co., engineers and contractors, where he served as an engineering superintendent and also as district manager at Detroit. In 1932 he organized the firm of Tom Brown and Associates, Detroit, air conditioning manufacturers representatives. After two years he returned to Chicago as general manager of the Autovent Fan and Blower Co.

In 1940 he joined the Avery Engineering Co., Cleveland, and shortly after that was hospitalized and for the past several years he had been in Eastlawn Sanitarium, Northville, Mich., and Veterans Hospital, Dayton.

Tom Brown was always active in Society activities serving as secretary of the Michigan Chapter in 1933-34 and president of the Illinois Chapter in 1939. He was a regular attendant at Society meetings and he had a fine reunion with many of his friends at the 49th Annual Meeting at Cincinnati, in January 1943. In that year he served as a member of the F. Paul Anderson Award Committee. He had a great interest in the Society and one of his letters reveals the pleasure he had in being a member and his appreciation of the interest of his host of friends.

"It has always been a great pleasure for me to put what time I could into Society work, as frankly, I have selfishly reaped a harvest of friendships that make up the greatest riches any one could possess. My good friends in the Illinois and Michigan Chapters, as well as many others in the Society, have done a great deal toward making my time go along happily with their good letters.

"I find that one lives from *mail to mail* and *visit to visit* and with these pleasant surprises the time really goes along quite rapidly.

"When one is taken out of circulation as long as I have been, it sure means a lot to know that one is not forgotten."

Mr. Brown was an enthusiastic golfer and in 1936 was the co-holder with R. T. Miller of the Society's Research Cup and in 1938 was the winner of the Research Cup and also a member of the Illinois team that won the Eichberg Memorial Cup.

Mr. Brown was also Past Commander of Winnetka, Illinois, Post No. 10, American Legion and was active in other fraternal organizations.

The Officers and Council extend their deepest sympathy to Mrs. Brown and to his sons who survive.

EQUIPMENT DEVELOPMENTS

For your convenience in obtaining more information about any of this equipment, see coupon on this page. Add the new products and companies listed here to your Directory Section which you received in your January 1946 Heating, Piping & Air Conditioning and thus keep your records of sources of supply up to date throughout the year. Single asterisk indicates equipment not listed in Directory Section; double asterisk equipment and manufacturer not listed.

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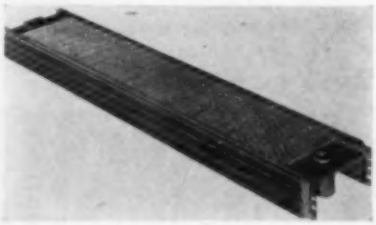
No. 3226—A new, completely nonferrous heating unit and additional enclosure types and sizes have been incorporated in the line of convector radiators offered by the Modine Mfg. Co., Racine, Wis.

Units are completely nonferrous throughout with headers, tubes, and fins of copper or copper alloy. Corrosion hazards resulting from electrolysis are thus said to be eliminated by avoiding the use of dissimilar metals. One piece construction achieved by brazing or otherwise metallically bonding all parts into an integral unit, together with rigid steel side plates, are said to have enabled these units to withstand severe battle shock aboard the navy's largest ships.



According to the manufacturer, an important feature is the universal header which permits the use of one style of heating unit on practically any two pipe steam or hot water piping arrangement. Also featured are adjustable brackets for obtaining correct pitch and elevation of the unit and a choice of two types of air chambers for use on hot water systems. The flush recessed style of enclosure, dropped during the war, has been reinstated and all recessed and floor cabinet enclosures are now available in a 16 in. height. In addition, the 3⅝ in. depth of convector has been included in the line.

Stainless Steel Welding Fittings

No. 3227—An extensive line of stainless steel welding fittings is now being manufactured by Tube Turns, Inc., 224 E. Broadway, Louisville 1, Ky., and is available through distributors located in all principal cities.

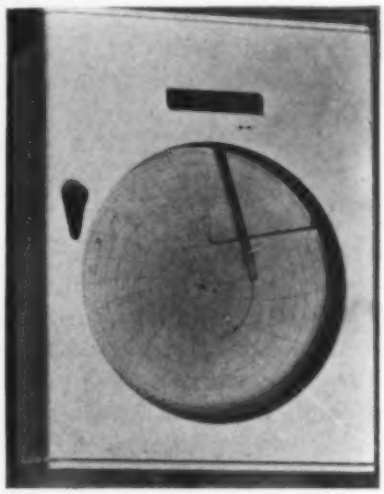
The line includes 180 deg long radius returns, 45 and 90 deg long radius elbows, straight tees, reducing outlet tees, caps, eccentric reducers, concentric reducers, lap joint stub ends, laterals and crosses—all in standard and extra heavy weights. Also included are welding neck, slip-on, threaded, blind, socket type, welding, reducing, orifice, and lap joint flanges. Sizes range from ¾ through 12 in.

These fittings may be obtained in three grades of stainless steel which the company considers will meet 95 per cent of industry's standard requirements. Outstanding advantages are said to be resistance to corrosion and strength and ductility at extreme temperatures.

Refrigerated Locker Plant Thermometer

No. 3228—Development of a new recording thermometer, designed specifically to meet the requirements of refrigerated locker plants, has been announced by the Minneapolis-Honeywell Regulator Co., 2604 4th Ave., S., Minneapolis 8, Minn.

The instrument covers a range from –30 F to 70 F and is equipped with a seven-day chart. Using a mercury filled thermal system, the new thermometer is said to permit evenly spaced chart graduations over its entire range, thus providing for easier and more accurate readings.



To comply with various state laws, the thermometer may be operated either by an electrical or hand wound clock mechanism. A smooth white enamel case houses the unit.

A reverse reading chart is optional for locker plant operators who desire temperature recordings nearer to the perimeter of the chart. This facilitates visual readings because the recorded temperature line for any given period of time covers a greater distance than a similar line drawn nearer to the center of the chart.

Industrial Oil Burner

No. 3229—A new "Arc-Heat" industrial oil burner has been announced by the American Heat Corp., Bay State Bldg., Lawrence, Mass.**

Having been tested in commercial installations for over a year, the burner is said to offer remarkable efficiency and to be completely unlike any other burner

[*Continued on page* 192]

Heating, Piping & Air Conditioning, [9-46]
6 N. Michigan Ave., Chicago 2, Ill.

Please ask the manufacturers to send me more information about the equipment mentioned under the following reference numbers in Equipment Developments and Recent Trade Literature.

(Circle each number in which you are interested)

3226	3227	3228	3229	3230	3231	3232	3233	3234	3235
3236	3237	3238	3239	3240	3241	3242	3243	3244	3245
3246	3247	3248	3249						
6269	6270	6271	6272	6273	6274	6275	6276	6277	6278
6279	6280	6281	6282	6283	6284	6285	6286	6287	6288
6289	6290	6291	6292						

Name..... **Title**.....

Company .....

Address .....

City..... **Zone**..... **State**.....

THEY KEEP

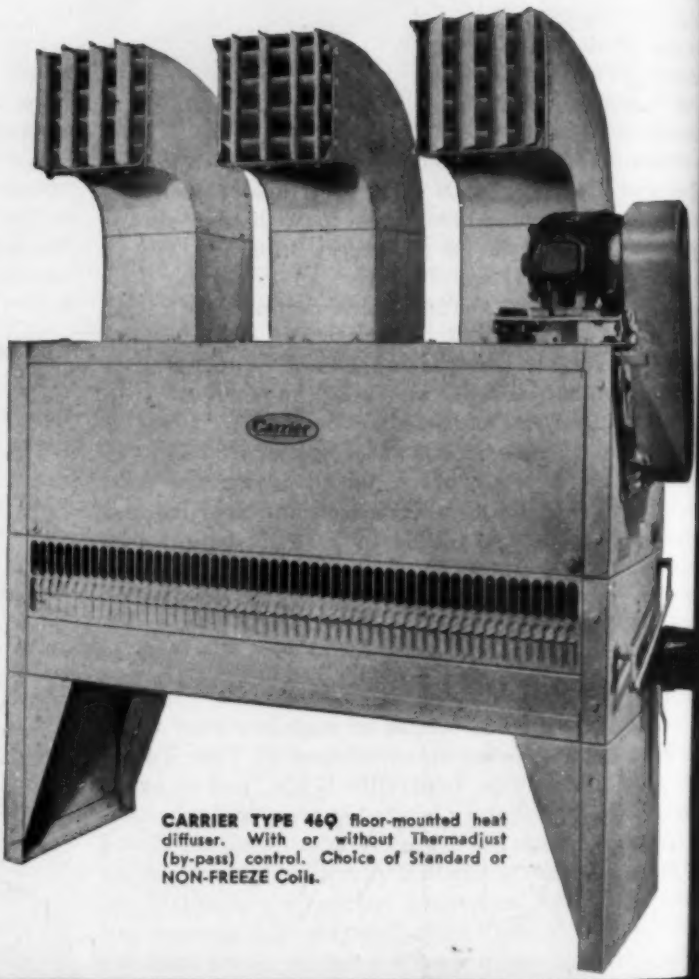
TEMPERATURE UP

FUEL BILLS DOWN

Heating large enclosed areas is simple and economical with Carrier Heat Diffusers. These units have no superior when it comes to hard, continuous service in factories, warehouses, garages and other industrial-type buildings. They deliver even heat at full or partial loads. Heat is controlled either by Carrier steam-distributing tube coils of the non-freeze type and modulating valves, or by Carrier's Thermadjust feature with modulating control at full steam pressure. No duct work is required.

Multiple discharge outlets and adjustable louvers permit air to be discharged in any direction—make these units adaptable to any type of space. They're adaptable also to ventilating and drying. One of several types is shown. Centrifugal-type fans are dynamically balanced. Belt drive, with vari-speed pulleys, provides a wide range of speeds and capacities. Design permits field conversion to floor or suspension mounting with vertical or horizontal discharge. Diffusers are shipped as a complete unit—ready to operate.

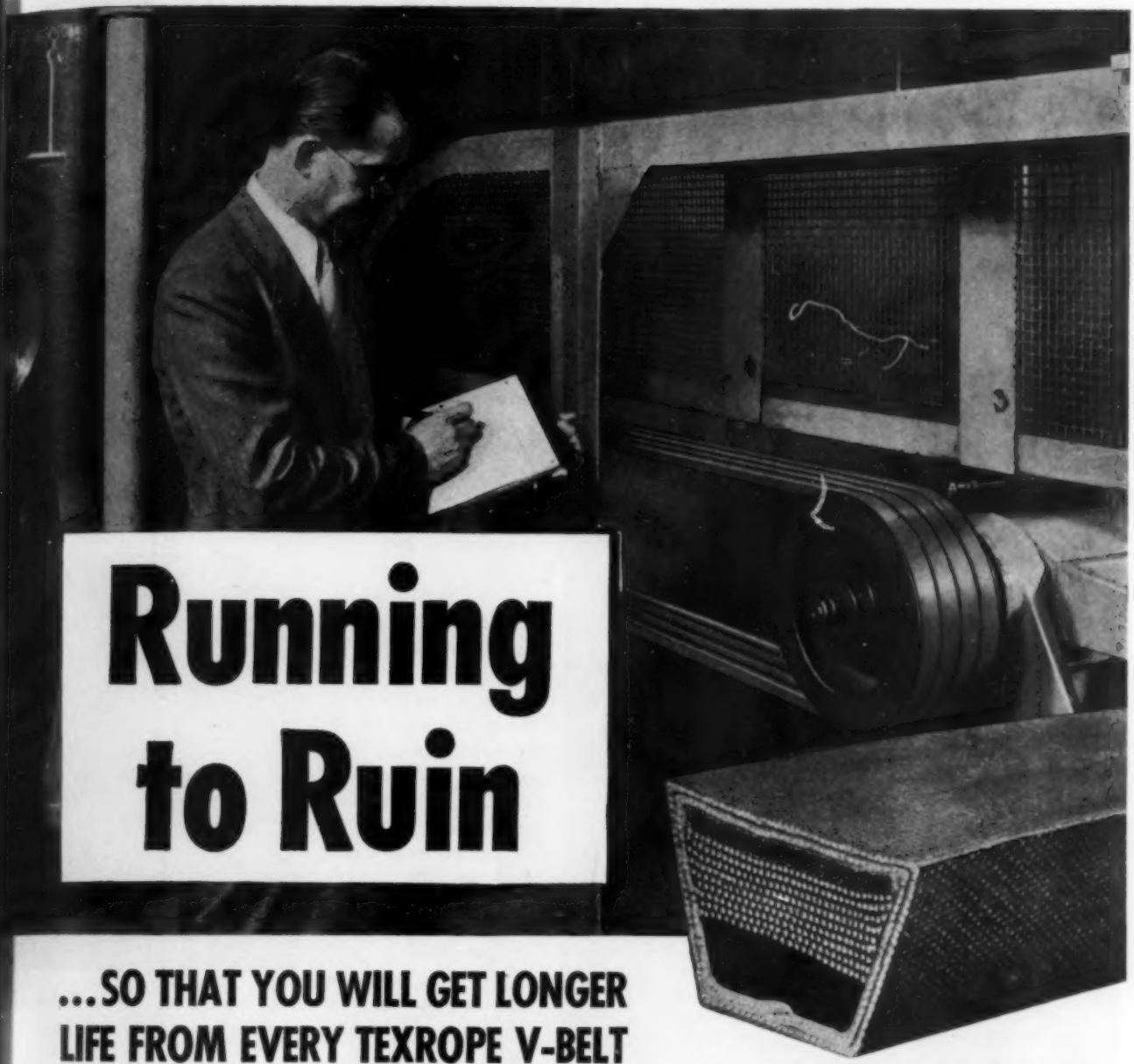
These heat diffusers are the product of Carrier leadership in engineering design and manufacturing skill. Carrier's long experience in industrial heating, refrigeration and air conditioning is your assurance of long, trouble-free service. Carrier Corporation, Syracuse, New York.



CARRIER TYPE 46Q floor-mounted heat diffuser. With or without Thermadjust (by-pass) control. Choice of Standard or NON-FREEZE Coils.

Carrier

AIR CONDITIONING
REFRIGERATION
INDUSTRIAL HEATING



Running to Ruin

...SO THAT YOU WILL GET LONGER LIFE FROM EVERY TEXROPE V-BELT

THIS MACHINE tests TEXROPE Super-7 V-Belts — runs them at varying speed and load until they give out. A trained operator records results: How long did they last? What broke down first: Cords? Cover? Rubber compound? Does destructive heat develop?

Out of tests like this have come stronger cords—tougher, more wear-resistant covers — resilient, cooler-running rubber — vital improvements in design and construction. TEXROPE Super-7 V-Belts are today the BEST in 20 years of V-Belt experience.

A-C FOR COMPLETE V-BELT SERVICE

Call your nearest Allis-Chalmers office or dealer for ALL your V-Belt drive needs — TEXROPE Super-7 V-belts all types, all sizes; TEXSTEEL, TEXDRIVE and "Magic-Grip" Sheaves; VARI-PITCH Sheaves and SPEED CHANGERS. They're all engineered and backed by the originators of the Multiple V-Belt Drive for industry. ALLIS-CHALMERS, MILWAUKEE 1, WIS.

A 2099

PICK YOUR SUPER-7 V-BELT

Heat-Resisting Super-7
Stands temperatures up to 180°. The TEXROPE V-Belt for most drives.

Oil-Resisting Super-7
Neoprene cover protects core against moderately oily or greasy conditions.

Oil-Proof Super-7
Made of Neoprene throughout. Use it when the belt must swim in oil.

Static-Resisting Super-7
Recommended where explosion hazard exists. Static-conducting element throughout cover won't wear off.

TEXROPE Super-7 V-Belts result from the co-operative research of two great companies—Allis-Chalmers and B. F. Goodrich—and are sold exclusively by A-C.

ALLIS CHALMERS

One of the Big 3 in Electric Power Equipment — Biggest of All in Range of Industrial Products

TEXROPE V-BELT DRIVES

ADSCO

CONDENSATE METERS

check SKYROCKETING fuel costs

• Today's soaring costs of fuel, labor and overhead make it more important than ever before to meter your steam consumption. Whether you generate your own steam, or buy steam, or sell steam, ADSCO Condensate Meters give you an accurate check on costs and losses.

Learn how ADSCO Meters pay their low cost by showing up fuel waste caused by steam line leaks, open doors or windows, and inefficient process operations. Write for Bulletin No. 35-80 HP.

AMERICAN DISTRICT STEAM CO.
NORTH TONAWANDA, N.Y.

(*Continued from page* 189)

on the market. Among the features which the manufacturer claims to be outstanding are (1) the use of a rarefaction chamber in which the oil is mixed with air or steam, is transformed into a gas, and is then sprayed into the combustion chamber, (2) the use of induced air in the fire box, eliminating the necessity for blowers, and (3) the use of a modulating valve designed to vary the amount of fuel being burned and also to regulate the amount of air accordingly.

The burners are now being manufactured for use in steam boilers, annealing ovens, melting furnaces, sand dryers, and in many other types of industrial and commercial equipment.

Room Air Conditioners

No. 3230—Three newly-created "Yorkaire" room conditioners, two of them window sill models, have been announced by the York Corp., Roosevelt Ave., York, Pa.

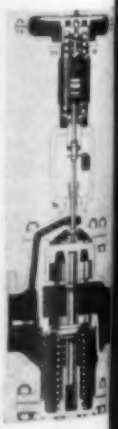
These new models, which may be used the year 'round, are designed to lower both temperature and humidity, clean and circulate the air, and remove stale inside air. They are said to be compact, easily installed, incorporate readily removable filter pads, and to be priced in the low and medium range.

Window type units are available in both the ½ and ¾ hp size and the console type (shown here) is offered in the ¾ hp. size. They are designed for conditioning rooms in homes, small offices, etc.

Diaphragm Regulating Valve

No. 3231—A new line of single-seated, balanced diaphragm regulating valves for process industries and power plants has been announced by the Leslie Co., 51 Delafield Ave., Lyndhurst, N. J.

The valves are made with cast iron, bronze, and steel bodies for steam, air or gas pressures up to 100 psi and are designed to be actuated by standard pressure or temperature instruments or controllers using 20 psi air or clean water.



Supplied as standard equipment with 800 Brinell hardened stainless steel main valves and stellited seat rings, these units are said to provide equal or greater capacities than most double-seated regulating valves of the same size, and have the additional advantage of being tight in dead-end service.

For temperatures over 450 F, steel diaphragm valves are equipped with a newly designed "Conden-Seal" cooling bonnet.

"Since 1890

Good people to do business with." That is a statement often made about Spencer. It's the best compliment we could ask for—and we promise to live up to it.

Spencer *knows* the heating business. Through the years our engineers have progressively modernized the Spencer line. Today's modern Spencer steel boiler is not just "another package" but a time-tested, *proven* product . . . a good reason why men who "live" heating problems recommend Spencer . . . why many of the most reliable distributors in the industry sell Spencer.

There's a precision-built Spencer unit for every individual heating job . . . every type of heat . . . every size of building . . . a premium boiler at no additional cost. Write or wire *now* for a catalog of the complete Spencer line.

SPENCER HEATER

Division—The Aviation Corp., Dept. H-8, Williamsport, Penn.

THE STOKER BUSINESS IS GOOD BUSINESS

. . . AND IT'S EVEN Better WHEN YOU'RE SELLING

FREEMAN LONG-LIFE STOKERS

FIRST CHOICE OF AMERICA'S FOREMOST HEATING ENGINEERS

IMMEDIATE DELIVERY *on all sizes!*

See how Freeman gives you *everything* you need to do a successful job of stoker selling!

- A complete line of domestic, commercial, and industrial models in capacities up to 1200 pounds per hour.
- Sound engineering backed by research and testing.
- Years of experience dating back to the pioneer days of the industry.
- The security and protection of a financially strong manufacturer who will always be in the stoker business to serve you.
- The unequalled assistance and cooperation of the foremost plumbing and heating distributors in the country.
- Aggressive promotion.

Now! Give yourself the competitive advantage of Freeman's established leadership. *It will mean greater stoker sales and profits for you . . .*

FREEMAN STOKER DIVISION
ILLINOIS IRON & BOLT CO.
918 S. MICHIGAN AVE., CHICAGO 5, ILLINOIS

Three-Way, Diaphragm Operated Control Valve

No. 3232—Hammel-Dahl Co., 243 Richmond St., Providence 3, R. I., is now producing a line of diaphragm operated control valves for flow diversion service.

These valves are of the three-way type, and the position of the diaphragm operated valve plug determines the diversion of flow between the two outlet ports from 0 to 100 per cent. Units are offered in sizes from ½ to 6 in. inclusive, in cast iron, steel, bronze, or in stainless steels available in cast form. The valves feature the company's usual construction of through-bolted bonnet and blind-headed flanges, all steel diaphragm superstructure, valve position indicator, two-ply neoprene diaphragm, and grease lubricator for the stuffing box gland. They are suitable for operation with standard 2 to 15 lb air output controllers or for controllers requiring special spring rangers. For high temperature service, airfin bonnets are available. Face-to-face dimensions are in accordance with ASA standards.

Geared Pipe Threader

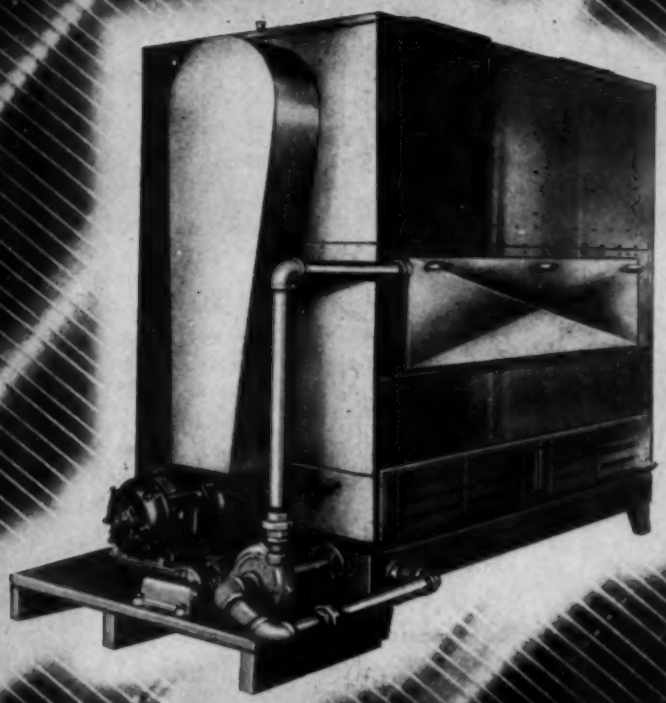
No. 3233—A quick-setting workholder and new balanced loop handles are two innovations on the new finished geared type threader announced by the Ridge Tool Co., Clark St., Elyria, Ohio.

The quick-setting workholder is said to afford an advantage in enabling a worker to set the threader exactly to size before being placed on the pipe; once on the pipe, only one screw needs to be tightened to hold it securely. All bushings have been eliminated. The balanced loop handles on the enclosed gear case are designed to prevent slipping or tipping over.

The manufacturer also claims economy of operation by virtue of a twin-anchored drive shaft that turns in oilless bronze bearings. The new unit is provided with four sets of five high speed steel dies to cut standard or tapered threads, drip threads, and short nipples. A ratchet handle is provided and a special design universal drive shaft is available for power drive.

Gasoline-Driven Power Unit

No. 3234—The "Power-Package," consisting of a 6 hp gasoline engine, automatic clutch and transmission, is a postwar product developed by Salsbury Motors, Inc., 4464 District Blvd., Los Angeles 11, Calif.



New!

COOLING TOWERS BY MARLO

- Triple-type — Induced Air / Wetted Surface / Water Spray
- Compact in space, weight, and price
- Outdoor—Indoor
  3 to 50 Tons—Built sectionally

Write for Bulletin 406

MARLO
HEAT TRANSFER SURFACE

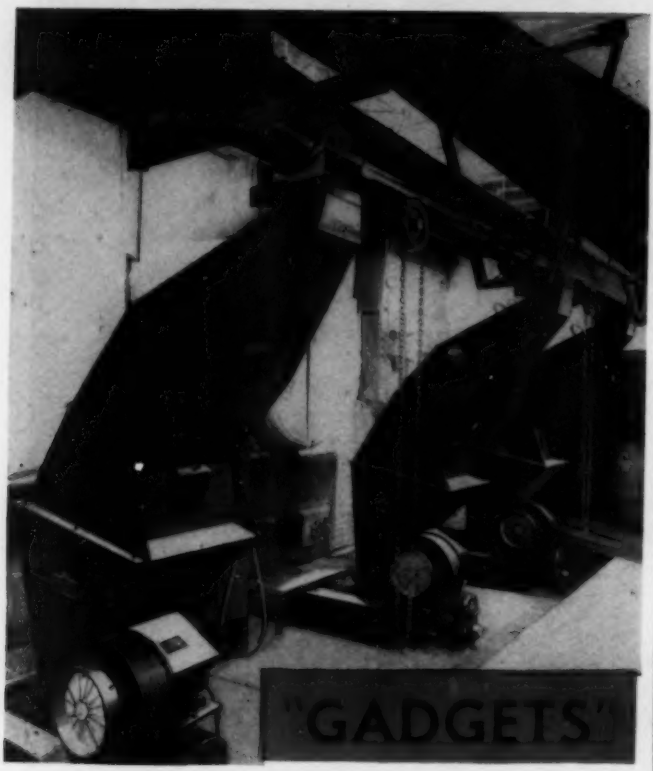
Ball-Bonded Blast Coils—Cooling and Heating • Air Conditioning and Refrigeration Apparatus • Industrial Blower Units • Unit Coolers Evaporative Condensers and Coolers • Low Temperature Apparatus

MARLO HEAT TRANSFER

See our new line in Cleveland

MARLO COIL COMPANY

ST. LOUIS 10, MISSOURI



help sell Link-Belt industrial stokers

Link-Belt has always followed the precept that there is more to business than just selling goods—and there is more to selling industrial stokers often than just installing the firing equipment.

In a large auditorium, three Link-Belt POWER-FLEX stokers solved not only the firing and combustion efficiency problems, but the flexibility requirements also . . . in combination with a Link-Belt continuous flow conveyor and screw conveyor with automatic limit controls . . . permitting the operator to spend most of his time outside the boiler room.

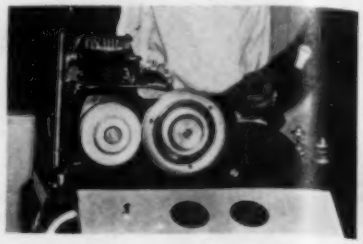
Engineering assistance is but one of the advantages of selling Link-Belt stokers. Let us tell you more about the Link-Belt franchise for both industrial and domestic stokers.

10422

LINK-BELT COMPANY
Stoker Division
2410 W. 18th St. Chicago 8, Illinois

LINK-BELT Magic Flame and Power-Flex STOKERS

According to the manufacturer, a wholly owned subsidiary of Northrop Aircraft, Inc., the unit will power light vehicles or fit almost any stationary use in which a small engine is required to transmit power gradually and at varying speeds to a driven machine. It is offered for use with machine tools, blowers, fans, compressors, stokers, pumps, and many other industrial machines.

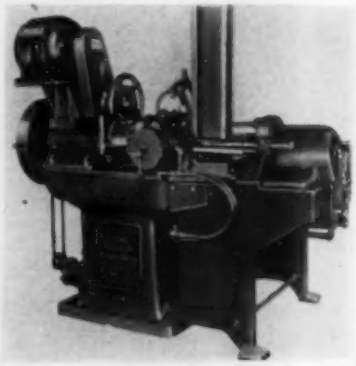


"Straight-shot" carburetion, an improved system of mixing and feeding fuel, is said to be a feature of the engine which weighs only 56 lb. The clutch and transmission are said to have fewer operating parts than any comparable equipment on the market. The clutch is engaged by centrifugal force and the transmission, which is fully automatic, utilizes centrifugal force to change the gear ratio over a wide range of speeds. Clutches, engines, and transmissions are available either separately or in the packaged unit.

Automatic Nipple Machine

No. 3235—The Landis Machine Co., Fifth and Church Sts., Waynesboro, Pa., has developed an automatic nipple machine for threading, reaming and chamfering both ends of pipe nipples.

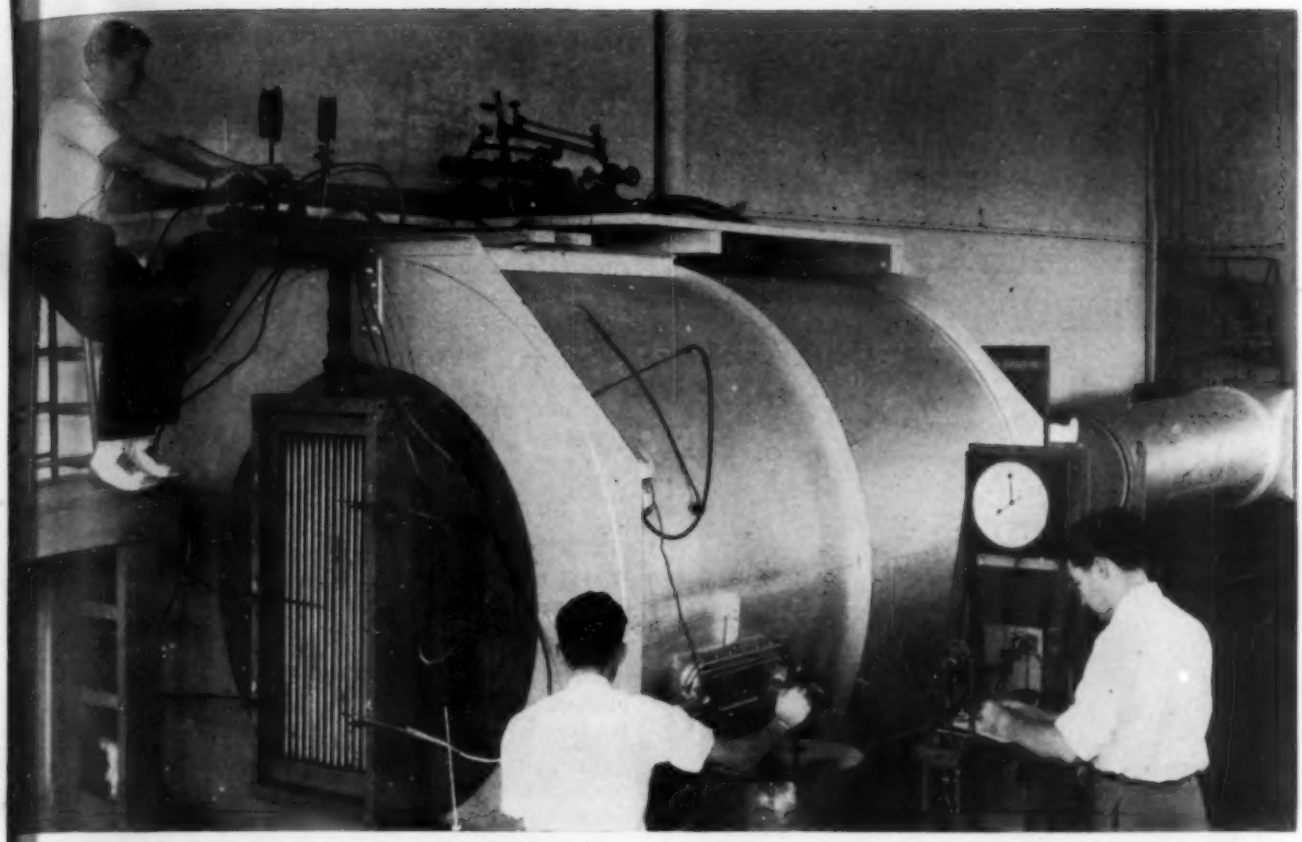
Machines are built in two sizes; the 1 in. machine for ½, ¾ or 1 in. pipe sizes and the 2 in. machine for 1, 1¼, 1½, or 2 in. sizes. The smaller machine can be equipped to handle any one diameter of pipe or combination equipment can be furnished for both ½ and ¾ in. pipe sizes. The larger machine can also be equipped to handle any one diameter of pipe or combination equipment can be supplied too to take 1 in. and 1¼ in. pipe sizes and 1½ and 2 in. sizes. The maximum length of nipples which can be cut is 6 in.



The machine has two spindles, each equipped with threading die head and reaming attachments. The two carriages have air operated vises for gripping the nipples and the left hand carriage carries an adjustable magazine from which the nipple blanks are fed automatically into the machine. Power is supplied by a constant speed motor with change gears provided to give the required spindle speeds for the pipe size to be threaded. A safety clutch is built into the drive to prevent serious damage in the event of a jam.

Swivel Pipe Coupling

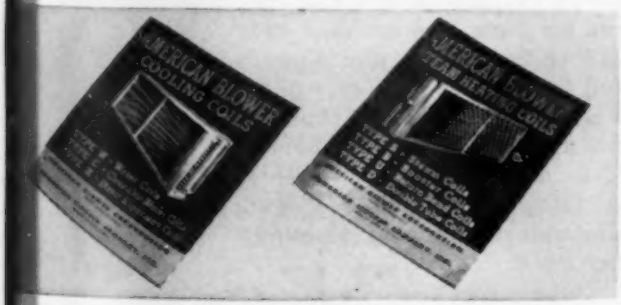
No. 3236—The "All-Flex" ball bearing, swivel pipe coupling offered by the Snyder Sales Corp., 5225 Wilshire Blvd., Los Angeles 36, Calif., has been designed to convey fluids under high pressure through a pipe which swivels or rotates a full 360 deg.**



Scene from American Blower laboratory showing one of the many tests run on heating and cooling coils.

What's been done to help you select the BEST in coils

Heating or cooling coils may look like a simple air, but actually hour upon hour of engineering es into every design. American Blower engineers ork constantly to bring you superior coils—coils at are right for the specific job you want done. he illustration shows only one of hundreds of sts. But now there is an easy way to get full nefits of *all* our research.



Here's how . . .

nd for these two bulletins. If you are ever faced ith the selection, sizing and installation of ating or cooling coils, these authoritative hand-books will save you many hours. Here in easy-to-use form is the information our engineers have compiled through lengthy research. We know these bulletins will be a big help to you in coil selection and application.

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For your copies, call or write the nearest American Blower Office. Request Bulletins B-1218 and 1521.

AMERICAN BLOWER

AMERICAN BLOWER CORP., DETROIT 32, MICH.

Division of American Radiator & Standard Sanitary corporation

When it's a *Gas* boiler job

. . . it pays to recommend a

Mueller Climatrol unit

You're sure of a satisfied customer—proud that you've given him the latest in gas boiler equipment. Completely redesigned and restyled — for better appearance, better performance. New, exceptionally quiet burner. Write today for "Facts and Figures" Bulletin F-320 giving full details.

L. J. Mueller Furnace Co.

2007 W. Oklahoma Avenue • Milwaukee 4, Wisconsin

MUELLER Climatrol

A combination of multiple synthetic packings and metallic seals are said to offer protection against leakage at high and low pressures, and a double row of ball bearings, together with metal-backed packings, are said to give low resistance to rotation.

This compact unit is made of steel and is available in sizes from ⅛ to 1 in. nominal tube diameter with a variety of threaded and elbow connections.

Test Kit for Boiler Feedwater Analyses

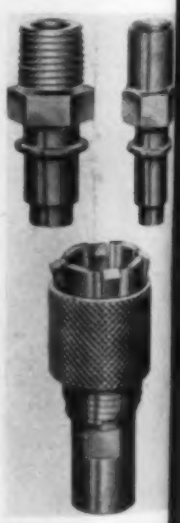
No. 3237—The testing of boiler feedwater for hardness, alkalinity and chloride content may be accomplished with a new boiler room test kit made by The Permutit Co., 330 W. 42nd St., New York 18, N. Y.

The kit, containing solutions and pipette droppers for accurate measurements, comes in a portable white enamel cabinet, 10 x 10 x 4¾ in. It can be modified to permit the testing of sodium sulfite in boiler water.

Air Hose Coupling

No. 3238*—A new air hose coupling, which the manufacturer guarantees to be leakproof under working pressures up to 1000 psi, is offered by the E. B. Wiggins Oil Tool Co., Inc., 3424 E. Olympic Blvd., Los Angeles 23, Calif.

Among the features which are said to afford savings in time, money, and materials are the speed and ease with which the unit may be connected and disconnected, and the ease and speed with which gaskets may be replaced. To connect or disconnect the coupling, a knurled ring is simply pulled down or pushed up.



Besides air, the device will handle oxygen, oil, aromatic fuel, and kerosene. The coupling body has ¼ in. female pipe threads and is available with or without a self sealing valve. A choice of two nipples is provided; one with ¼ in. male and the other with ¼ in. female pipe threads. The entire unit is made of aluminum bar stock.

Flexible Gear Box and Universal Joint

No. 3239—Brooks Equipment Co., 217 Hudson St., Hoboken, N. J., has introduced a new flexible gear turret and a new line of universal joints.

The gear turret, shown here, is designed for

Unless you are pretty sharp-eyed, you'll pass by "old man walking stick" as just another dead twig on a tree.

And there are a lot of other things that look different from what they really are . . . for example, certain cut-rate plumbing supplies. The wise plumber will avoid the pitfalls that come with inferior merchandise by relying on the reputable jobber in his vicinity.

Dollar for dollar, the trade-marked, quality merchandise he stocks is your best plumbing buy.

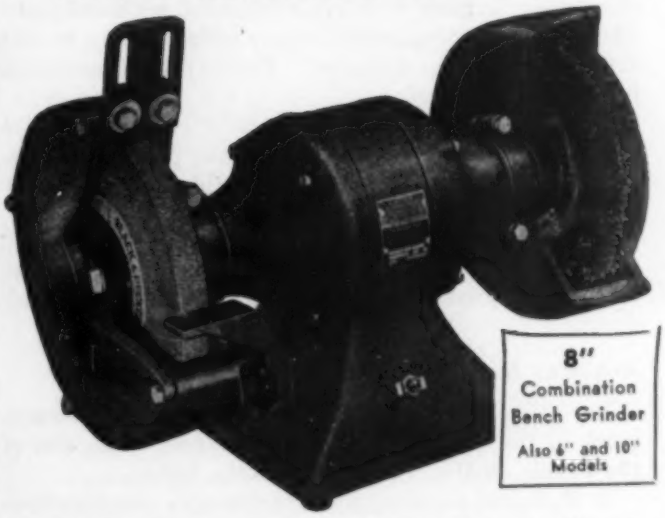
KUHNS BROTHERS CO.
DAYTON 1, OHIO

K
CAST FITTINGS

"K" fittings are carried in stock at Malleable Iron Fittings Co., Branford, Conn.; M. I. F. stocks at Kuhns Bros. Co., Dayton, Ohio.

Sharper TOOLS! Smoother JOBS!

. . . with Black & Decker BENCH GRINDERS



Black & Decker Bench Grinders keep a keen cutting edge on your tools—help you turn out cleaner work, faster. These Grinders handle many other jobs too! They grind pipe, rings, connections, fixtures, etc.—wire brush, buff, burnish, polish and finish sheet metal—remove weld marks and saw or shear burrs.

Long wheel spindles and tapered end housings make room for large or odd-shaped pieces. Powerful motors maintain high operating speeds under pressure—help you get work done sooner. Many other features.

Ask your Black & Decker Distributor about Black & Decker Bench Grinders and other Electric Tools for heating, piping and air-conditioning work. Or write to: The Black & Decker Mfg. Co., 628 Pennsylvania Ave., Towson 4, Maryland.

"WHIRLWIND"* WIRE BRUSHES

Perfect team-mates for Black & Decker Grinders. Tougher, denser, last longer . . . because each tuft is locked in place to stay. Six wheel sizes: 4" to 12". Three wire sizes. Three thicknesses. Fit almost any arbor or spindle.

*Trade Mark Reg. U. S. Pat. Off.

LEADING DISTRIBUTORS EVERYWHERE SELL

Black & Decker

PORTABLE ELECTRIC TOOLS

DRILLS • HOLE SAWS • DRILL STANDS • PORTO-SHEARS
BENCH GRINDERS • SANDERS • PORTABLE GRINDERS

necting shafts which form any angle between 0 deg (straight line) and 95 deg on either side of zero angle. Bevel gears are attached to shafts which are supported on hinged housings. This is said to permit the shafts to oscillate and rotate simultaneously. Horsepower characteristics are based on the Lewis formula and are consistent with standard power transmission practice for shafts ranging in size from ¾ to 1½ in. diameter. Two types of mounting brackets are offered.

The new universal joint is said to employ a device which provides a positive lock for the pins and a convenient method of disassembly. Heat treated alloy steel is used throughout.

A. J. Quinn, newly elected president, has announced the consolidation of the company's executive and sales offices and production facilities at the Hoboken address.

Acquires Manufacturing Plants

No. 3240—M. B. Hott has acquired all of the common stock of the Alliance Mfg. Co., Alliance, Ohio, and of the Precision Mfg. Co., of Bergholtz, Ohio.

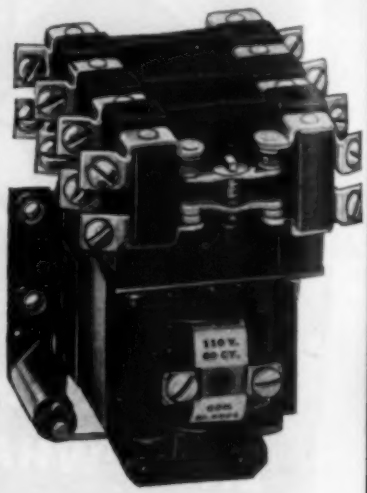
R. F. Doyle, formerly general manager and treasurer of Alliance, is president of both companies, for many years the world's largest producer of fractional horsepower electric motors in special types which are used by the radio and electronic control industries.

According to Mr. Doyle, the company will continue the policies and relations with its customers and suppliers which have been consistent with its operation over the past 21 years.

Solenoid Relay

No. 3241—Unusual flexibility is said to be the outstanding feature of a new line of universal solenoid relays developed by the Allen-Bradley Co., 1311 S. First St., Milwaukee 4, Wis.

In these relays, which are in addition to the regular Bulletin 700 line of solenoid relays, a simple shift of connections makes it possible to change a set of normally open contacts to normally closed operation or vice versa. This universal type, which should be used only for switching circuits of like polarity and which cannot be used as reversing switches, can be obtained in either the open style or mounted in an enclosure. Three types of enclosures are available.



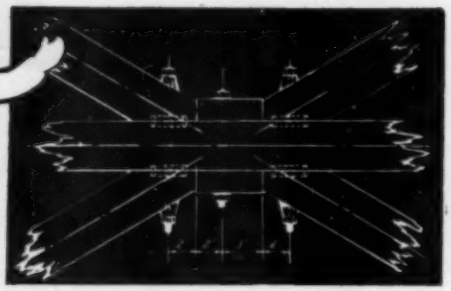
. . . . . how BIG is your FURNACE DOLLAR?
Consumers, dealers and jobbers all agree that . . . . .
SYNCROMATIC
offers the biggest dollar's worth of FURNACE SATISFACTION on the market today in . . . .
Beauty, Performance, Economy, Safety, Satisfaction!
1946
COAL
GAS
OIL
Syncromatic Corporation
WATERTOWN, WISCONSIN

L-H Towers withstand wind pressures up to 150 m.p.h. without guy lines.

ANOTHER L-H *Feature* MAXIMUM STRUCTURAL STRENGTH

100% WORKING STRESS OF MEMBERS

1. Teco split-ring timber connectors used at all stress joints.
2. Bolt-thru bracing connected by sidehill washers (no notching).
3. Unit stresses of U. S. Forest Products Laboratory Handbook incorporated in all L-H designs.
4. Prefabricated of heart Redwood, or Tidewater Red Cypress.



L-H FEATURES: • Low pumping and power cost... open-flume and gravity-type distribution. • Maintenance-free reduction gear and drive shaft. • Pre-tested performance based on L-H 48-year installations record.

Write for Catalog 440-A.

Lilie-Hoffmann
COOLING TOWERS, INC.
Manufacturers of All Types of Water Cooling Towers
GENERAL OFFICE: 4239 DUNCAN AVE., ST. LOUIS 10, MO.
PLANTS AT ST. LOUIS, MO., AND PLAINVIEW, TEXAS

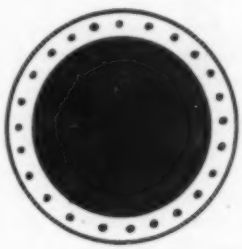
Avoid Waste *use* ROLLED CHROME LOCK GASKETING

NO need to gasket the hole when you use Chrome Lock Rolled Gasketing. Just finger press it in place. Its adhesive back stays put when and where you want it! No lace, tie, cement or tape to bother with.

Chrome Lock Gasketing is anti-wicking—anti-corrosive. Impregnated with a chromate pigmented compound, it is flame resistant.

Chrome Lock holds up under pressure of 500 lbs. per square inch at normal temperatures. It is not effected by oil, gasoline or salt. Pliable, it holds its plasticity indefinitely. It can be laminated, or lapped.

WHY PAY FOR THE HOLE?

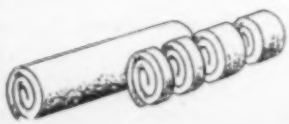


Use Chrome Lock Gasketing for piping, air conditioning, heating and refrigeration work. Write for FREE SAMPLE.

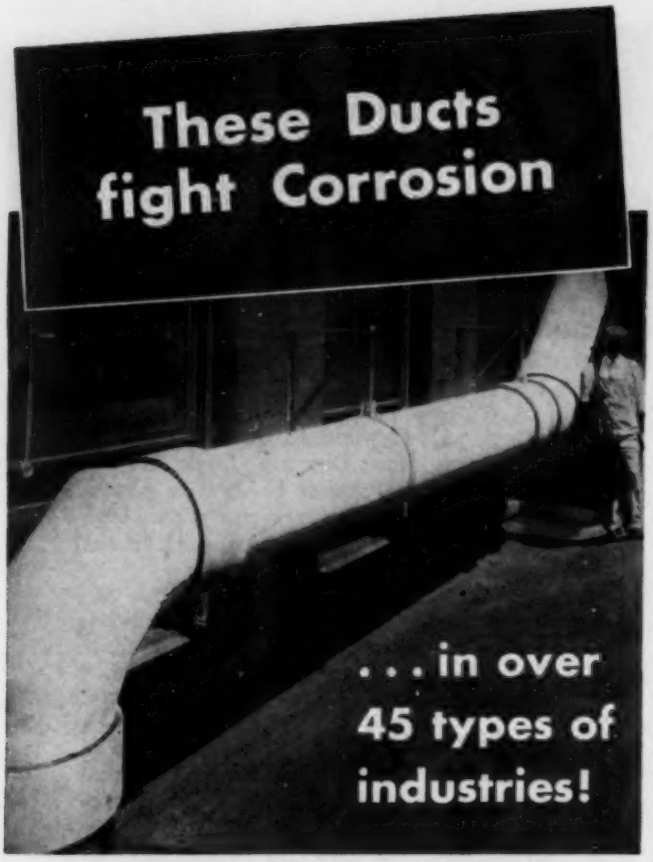
CHROME LOCK GASKETING

Available Now

Rolls **Slit Rolls**



Chrome Lock Gasketing
Chrome Lock
PRODUCTS RESEARCH CO.
634 S. WESTERN AVE., LOS ANGELES 5, CALIF.



FOR many years, ducts, vents and stacks of Transite Industrial Vent Pipe have helped solve corrosion problems in more than 45 types of industries!

This strong, durable pipe successfully resists many of the corrosive fumes, vapors, dusts and gases encountered in industrial operations . . . providing effective and economical venting service.

Transite Industrial Vent Pipe is rustproof . . . highly weather-resistant . . . needs no paint. Made of asbestos and cement, it is light in weight . . . easy and economical to install. Available in sizes up to 36" in diameter, with a complete line of Transite fittings to assure corrosion-resistance throughout the system.

For full details, Data Sheet DS-336 is free on request. Write Johns-Manville, Box 290, New York 16, New York.

JM PRODUCTS

Typical Industries which use Transite Industrial Vent Pipe

Aircraft	Foundry	Quarrying
Automobile	Furnace	Railroad
Baking	Furniture	Rayon
Bleaching	Gas	Refrigeration
Boiler Works	Glass	Rubber
Brewing	Laboratory	Sewage Works
Canning	Laundry	Shipbuilding
Ceramic	Leather	Shoe
Chemical	Meat Packing	Smelting
Dairy	Metal	Soap
Drug	Mining	Soft Drink
Electrical	Paint	Sugar Refining
Explosive	Petroleum	Textile
Farm Machinery	Potash	Tool
Food	Pulp & Paper	Water Works

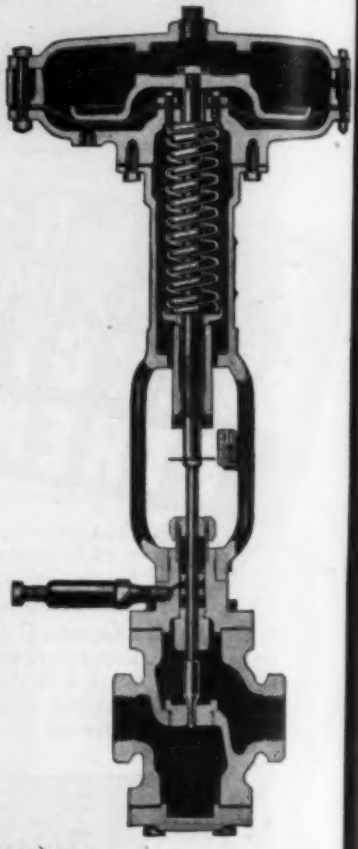
Johns-Manville
TRANSITE Industrial Vent PIPE

In general, the construction of this line follows that of the regular Bulletin 77 line. Double break, silver alloy contacts are incorporated and only one moving part is used. Relays are available in 2, 4, 6, and 8 pole construction with a maximum current rating of 10 amp, noninductive, and can be obtained for operation on 25, 30, 40, 50, and 60 cycles with various voltages from 6 to 660.

Reverse Acting Diaphragm Motor for Control Valves

No. 3242—Illustrated here is a new reverse acting diaphragm motor for control valves, type 567, offered by the Fisher Governor Co., 205 S. First Ave., Marshalltown, Iowa.

On this new top work, the sealing diaphragm construction is designed to provide reverse action without use of a stuffing box. Other features include large diaphragm area, long stroke action, enclosed spring, separate from pressure chamber, linear travel relationship to air pressure change, interchangeable with the direct acting, type T top.



Intended application of this diaphragm motor is for pneumatic operated control valves where push down to close inner valve construction is desirable, yet operation requires valve air opened, spring closed (closing on operating medium failure).

For field and general service, this top work allows the integral top mounting of the company's pressure pilot or time cycle intermitter directly on the valve. Pilot diaphragm assembly of integrally mounted unit is interchangeable with the upper diaphragm casing of this new top work.

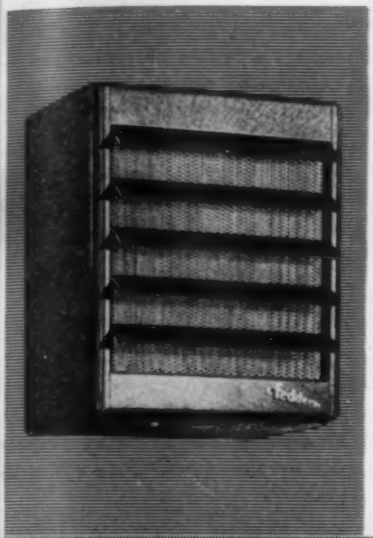
Air Conditioning Equipment Manufacturer in New Plant

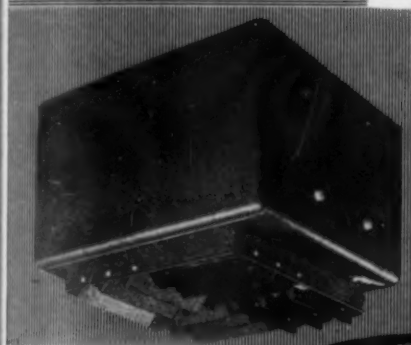
No. 3243—The United States Air Conditioning Corp. is now operating in a new plant recently purchased. The new building, at 33rd and Como Ave., S. E., Minneapolis, Minn., provides approximately 100,000 sq ft of manufacturing and office space.

The company designs and manufactures a line of blowers, fans, air washers, cooling and heating coils, humidifying and dehumidifying equipment, together with packaged air conditioning units.

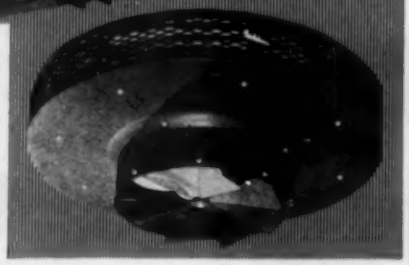
fedders HORIZONTAL DOWNBLOW Unit Heaters

Fedders Series 15 Horizontal Type Unit Heaters built in 100 to 1000 EDR capacities.





Fedders Series 12 Downblow Unit Heater at left. Used for medium ceiling heights of 15 to 25 ft.



Series 12 Downblow Models shown at right for 20-45 ft. ceiling heights.

The unit heaters illustrated are typical of the wide range of horizontal and downblow types for normal and high ceiling installations. They are designed to produce comfortable heating conditions, resulting from proper final temperatures, maximum air volume and a great degree of quietness. Write for catalogs.

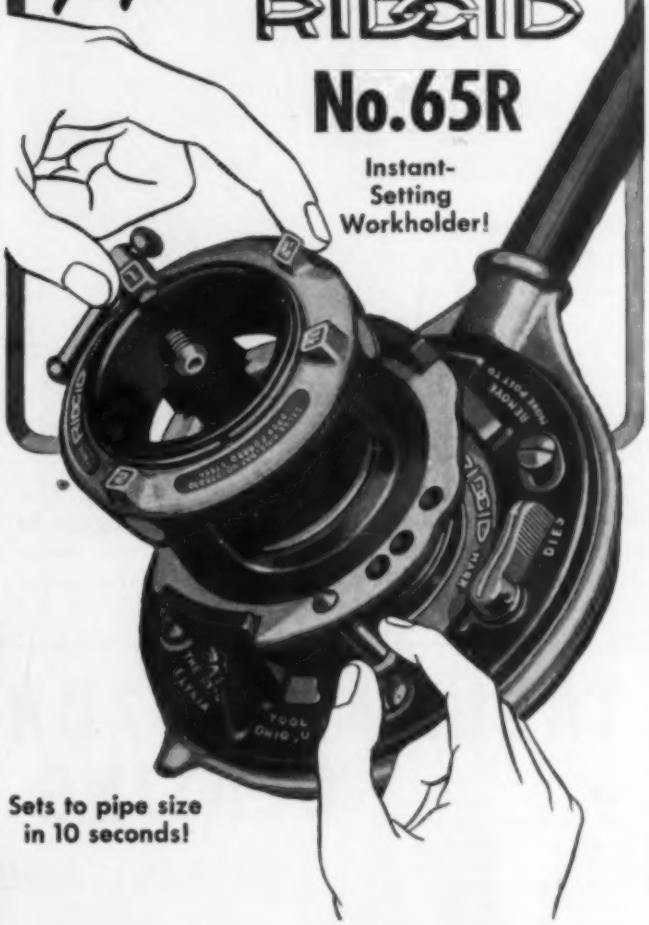
FEDDERS-QUIGAN CORPORATION

BUFFALO 7, N. Y. *Heat Transfer Specialists Since 1896*

Gives you a new idea of quick easy *pipe threading*

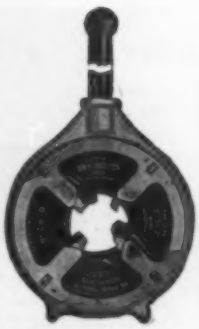
RIDGID No. 65R

Instant-Setting Workholder!



Sets to pipe size in 10 seconds!

This popular No. 65R cuts perfect threads on 4 sizes of pipe, 1" to 2," with 1 set of high-speed steel chasers — no changing, no loose dies to carry or lose. And it sets to pipe size in a mere 10 seconds! A turn of the cam plate sets the workholder to size instantly . . . one screw to tighten on pipe, no bushings to bother with. Many other worksaver features you like in this rugged precision-made tool. See No. 65R at your Supply House.



RIDGID *No. 65R stands up handily on its own feet.*

RIDGID

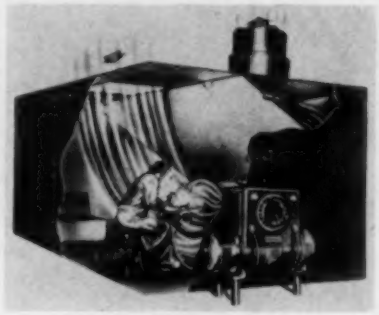
WORK-SAVER PIPE TOOLS

THE RIDGE TOOL COMPANY • ELYRIA, OHIO, U. S. A.

Production Welding Booth

No. 3244—A ventilated welding booth, designed to protect their health and provide more pleasant working conditions for production welders, is being planned for future production by the Hobart Brothers Co., Troy 1, Ohio.

The booth will be 9½ x 9½ x 7 ft high and is to be constructed of fabricated panels (16 ga sheet steel) and formed with companion flanges punched on 12 in. centers for bolting assembly. It is said this type of construction facilitates moving the unit to another location if desired. The door is of the sliding type with an overhead track and rollers. An exhaust fan draws air through an opening in the top and is used to discharge heat and welding fumes to the outside of the building.



New Company to Manufacture Thermostats

No. 3245—Formation of the Stevens Mfg. Co., Richfield Trust Bldg., Mansfield, Ohio, to engage in the design, manufacture and sale of electrical appliance and industrial thermostats has been announced by W. C. Stevens, president.**

Large scale production is now under way on several thermostat models with others to be announced in the near future.

Mr. Stevens was affiliated with the Westinghouse Electric Corp. for 21 years and at the time of his resignation was manager of thermostat sales.

Dial Type Pressure Gage

No. 3246—The "Supergauge," designed for heavy duty industrial service, is a pressure gage of large dial size offered by the United States Gauge Div., American Machine and Metals, Inc., Sellersville, Pa.

According to the manufacturer, internal parts are made of noncorrosive metals. Working elements are designed to withstand repeated pulsation and vibration, and the unit is styled so that it can be easily read at great distances. Gages are made for pressures from 30 to 10,000 psi and come in 4½, 6, and 8½ in. dial sizes. They are offered for use on boilers, diesel engines, hydraulic equipment, compressors, pumps, and for many other applications.

Air-Acetylene Attachment for Welding Torch

No. 3247—An attachment, designed to adapt any standard oxyacetylene welding torch for body tinning, soldering, and light brazing with an acetylene-air flame, has been developed by the Acet-A-Tip Co., 5069 W. Madison St., Chicago 44, Ill.**

The tips, which are available in three sizes and are machined from solid brass bar stock, are attached to

TALLMADGE ZONED CONTROLLED HEATING SYSTEMS

For better heating at less cost

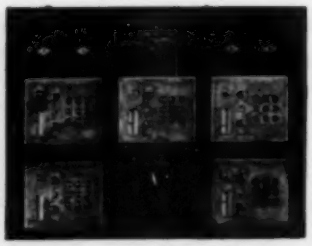


Figure 1

Figure 3

Figure 4

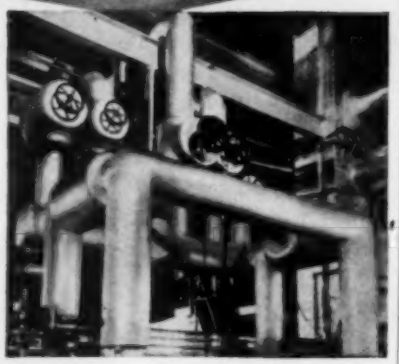
Figure 2

Figure 5

THE TALLMADGE equipment shown above is for a Zoned Heating System, either automatic or remote control (Figure 1). Airotherms, (Figure 2) for outside control, determine the heating requirements and set Pressure Boxes (Figure 3) to regulate the steam pressure to each zone. Zone valves (Figure 4) control the steam passing to the radiators. Orifices (Figure 5) meter the steam entering the radiators.

ADVANTAGES: Precise control of steam. Elimination of over-heating and costly waste of fuel. Low cost of maintenance. Saving on outlay for original equipment. Greater comfort to occupants due to steady heat and modulated even distribution.

WEBSTER TALLMADGE & CO., Inc. 364 Glenwood Ave., East Orange, N. J.



In San Francisco, the foremost insulation firm is Van Arsdale-Harris, Novoid contractor, which employs a construction staff of 75 and maintains five warehouses. Above is the exterior of the new Acme Brewery, which this firm insulated throughout with Novoid Corkboard. At right, a typical Novoid cork pipe covering job installed by Van Arsdale-Harris, in the Merchants Ice & Cold Storage Co. plant.

All over America leading Insulation Contractors insfall NOVOID

YOU NEEDN'T WAIT to get delivery of Novoid Cork Pipe Covering. Well-known independent insulation contractors all over the country now carry warehouse stocks of many sizes and thicknesses.

Novoid Covering is the moulded rigid insulation that efficiently reduces your cost of refrigeration. It snugly hugs pipes and fittings of every size and shape, does not sag, does not absorb moisture or odors, and is long-lasting.

For complete facts about Novoid Corkboard, Pipe Covering, and sundries specially developed to make low-temperature insulations most efficient, write to Cork Import Corporation, 330 West 42nd Street, New York City.

NOVOID INSULATION

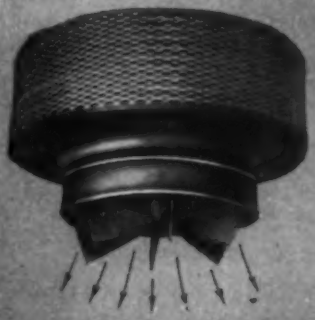
CORKBOARD · PIPE COVERING

McCORD HEATING AND AIR CONDITIONING PRODUCTS

BLAST HEATING COILS—AIR CONDITIONING COILS—ALL TYPES

All copper spiral fin tube construction with the tubes formed or bent permits uniform expansion without strain. Headers are heavy wall construction with tubes brazed into extended ferrules formed in the headers. Fins are helically wound and metallically bonded. May we send catalog?

VERTICAL TYPE UNIT HEATERS



A proven heating unit, modern in design, incorporating every unit heater advancement. McCord copper spiral fin tube construction creates air turbulence without undue restriction. Improved fan blades deliver large volume of air with minimum air noise. Guaranteed for 150 lbs. saturated steam pressure. Specify McCord.

HORIZONTAL TYPE UNIT HEATERS

A time proven heating unit, modern streamlined in design, quiet, yet its rugged construction adapts it to all types of installations, guaranteed for use on No. 150 p. s. i. steam pressure. All copper heating element, individual spiral fin tubes free to expand without strain on joints. Standard motors with standard bases used. Wide range of capacities.

Save fuel with McCord Unit Heaters.

McCord CORPORATION
DETROIT 11, MICH.

UNSURPASSED VALVES

K-10 SERIES

ELECTRO MAGNETIC LEVER VALVES

FOR CONTROLLING air, water, gas, steam, light and heavy oils, and refrigerants, etc. Suitable for pressures up to 1,000 pounds and temperatures up to 400°F.

Internal valve assembly can be removed for cleaning and inspection if necessary. Two-wire—A. C. or D. C.—current failure—packless construction—bronze body—stainless steel and bronze internal parts—soft seats available.

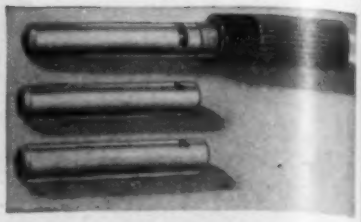
SIX TIMES THE POWER of ordinary solenoid valves. This feature assures high seating pressure and overcomes the effect of viscous fluids, such as heavy fuel oil. Adequate power of the solenoid insures positive operation. Available in normally closed or open types.

FOR COMPLETE SPECIFICATIONS on temperature, pressure and flow controls for industrial applications write for new 1946 Catalog 52C or contact Factory Branch or Distributor near you.

GENERAL CONTROLS
801 ALLEN AVENUE GLENDALE 1, CALIF.

FACTORY BRANCHES: PHILADELPHIA • ATLANTA • BOSTON • CHICAGO • DALLAS KANSAS CITY • NEW YORK • DENVER • DETROIT • CLEVELAND • PITTSBURGH HOUSTON • SEATTLE • SAN FRANCISCO • DISTRIBUTORS IN PRINCIPAL CITIES

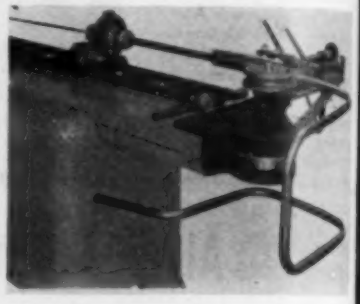
the welding torch by means of a base which is fitted with a 2 in. length of two-ply hose. The hose is forced over the welding tip torch. In using the device, the acetylene pressure is set at approximately 5 psi and the flame is adjusted by the torch valve. Oxygen is not used with these tips.



Hydraulic Tubing Bender

No. 3248—From one to 10 bends in a single length of tubing may be made with only one setting on the "Bendmaster" unit made by the Leonard Precision Products Co., 1100 Larson Ave., Garden Grove, Calif.

The new model, with a capacity of ⅜ to 1½ in. ferrous or nonferrous tubing, is hydraulically operated. It is designed for any production bending operation requiring multiple bends per tube and duplicate parts and is said to be especially suitable for use in the manufacturing of air conditioning and refrigeration equipment.



Oil Tank Gage

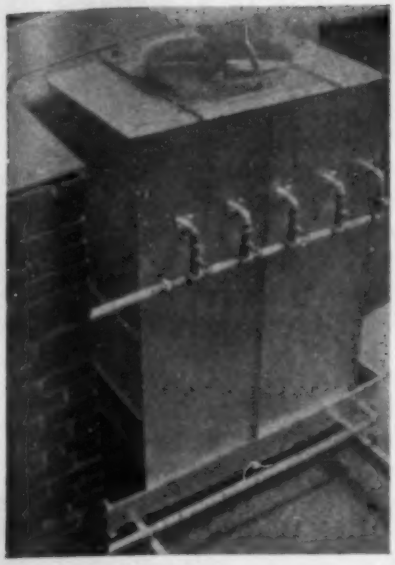
No. 3249—The Krueger Sentry Gauge Co., 1056 W. Mason St., Green Bay, Wis., has announced the new model "At-A-Glance" oil gage, developed for oil tanks for either 1½ or 2 in. opening.

It is designed for basement tanks of all sizes, and can be read from any direction. The reading scale is encompassed between two circular domes, providing four reading windows. Listed as standard by Underwriters Laboratories, Inc., the unit incorporates a simple lever action that moves the indicator disc up or down as the content of the tank raises or lowers. The inner dome is made of a hard, annealed glass and the outer dome is constructed of plastic material, cushioned at the base with gaskets.

OHI Plans Meeting

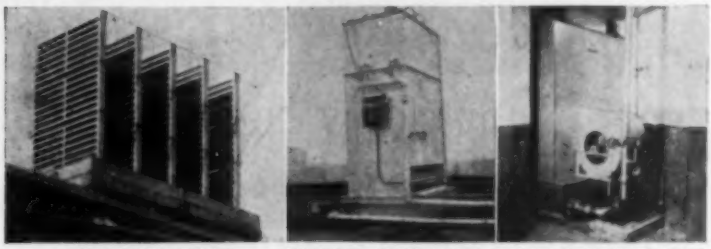
The board of directors of the National Oil Heat Institute, 30 Rockefeller Plaza, New York City, has voted unanimously to hold a convention and exposition in the spring of 1947. It is expected that the convention will be held in Boston.

TRAY COOLING TOWERS



COOLING TOWER ABOVE—40 Tons Refrigeration Capacity. 120 GPM. Induced Draft—Outdoor Unit.

SAVE 98% of CONDENSER WATER COSTS wasted thru REFRIGERATION and AIR CONDITIONING PLANTS



ATMOSPHERIC UNIT WORCESTER, MASS. | INDUCED DRAFT COOLING TOWER, PHILA., PA. | INDOOR COOLING TOWER, BROOKLINE, MASS.

TRAY COOLING TOWERS *offer*

★ ALL STEEL CONSTRUCTION
★ PACKAGE UNITS—REQUIRE LESS SPACE
★ OPEN DISTRIBUTOR HEADER — NO NOZZLES requires less horsepower for recirculating pumps.
★ INDOOR and OUTDOOR UNITS

Send for BULLETIN 46

Some open territory for factory representatives

NEW ENGLAND COOLING TOWER CO. 89 Broad Street BOSTON 10, MASS.

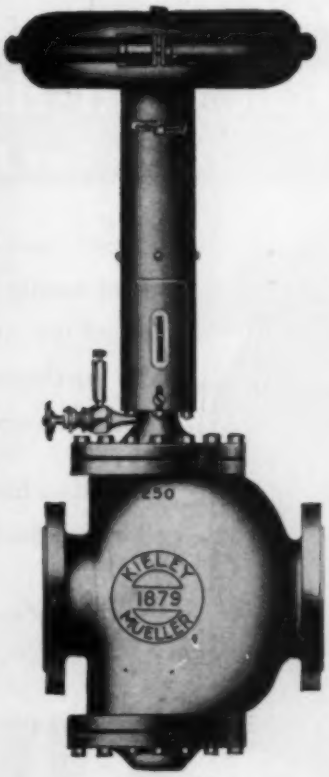
SOMETHING NEW HAS BEEN ADDED

Write for new Kontrol Motor bulletin. Send us your control problems.

KIELEY & MUELLER, INC., North Bergen, N. J.

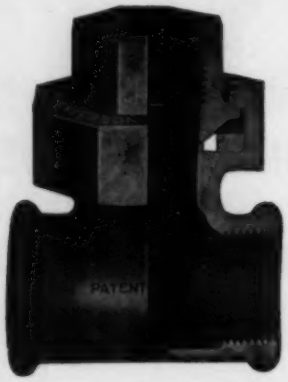
KONTROL KIELEY & MUELLER MOTOR

SEE OUR EXHIBITS—
BOOTH 155, NAT. CHEMICAL EXPOSITION
BOOTH 158, INST. SOCIETY OF AMERICA



- ***All Steel Diaphragm Motor Unit...*** lighter, tougher, stronger, more durable
- ***Boltless "Duoseal" Diaphragm Casing...*** tight sealing, quick acting clamp ring; molded highly flexible neoprene diaphragm
- ***Steel Tubular Yoke...*** enclosing long, heat-treated, cadmium plated spring; packing gland and spring adjustment accessible through door
- ***Valve Position Indicator...*** standard equipment; no extra charge
- ***High Capacity Valve Bodies...*** unrestricted flow areas, smooth flow; available in bronze, semi-steel, cast steel, and special alloys
- ***Superfinished Disc Guides...*** top and bottom guided Bevel, Percentage V Port, and Parabolic discs; minimum friction, increased life
- ***Modern Design...*** to meet modern process requirements for better control

THE UNIONS WITH THE RECESSED BRASS SEAT



JEFFERSON SPECIALTY UNIONS

Effect Big Savings in Labor

Jefferson Union Tees with union on the outlet assure labor savings and better piping jobs because they take the place of an all-female union, two nipples and an ordinary tee. Three joints are thus saved. Available in All-Female and Male and Female threads.

The brass seating in these and all Jefferson Specialty Unions is an exclusive feature which, located in a recess away from the runway of the fitting, leaves a straight bore assuring unrestricted flow in the pipe.

JEFFERSON UNION CO.

601 West 26th Street, New York 1, N. Y.

Factories at

Lexington 73, Mass. Lockport, N. Y.

RECENT TRADE LITERATURE

For your convenience in obtaining copies of these bulletins, see coupon on page 189. If you write direct to the manufacturer, describe carefully what literature you want, as the number given first in each item is for use only when sending requests to Heating, Piping & Air Conditioning.

Air Conditioning

No. 6269—"Kathabar" air conditioning systems, using a concentrated solution of lithium chloride to accomplish dehumidification, are discussed from the standpoint of operating experience in a number of installations in the Vol. 1, No. 2 issue of The Moisture Engineer, a house organ of Surface Combustion Corp., 2375 Dorr St., Toledo 1, Ohio.

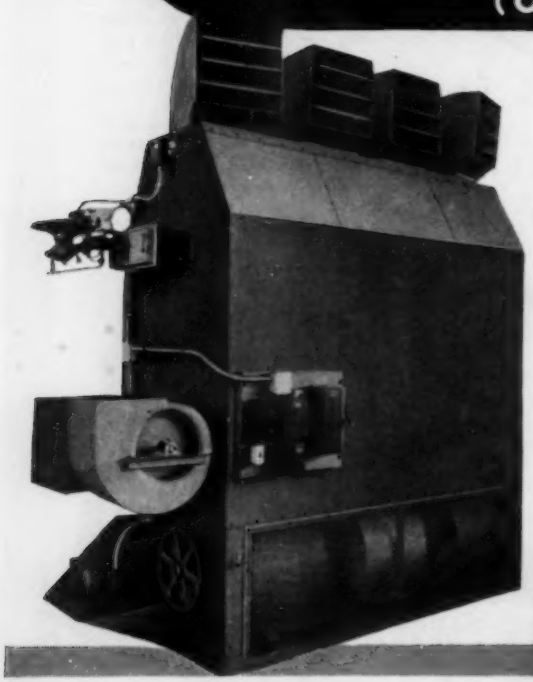
Automatic Defrosting in Low Temperature Refrigeration Systems

No. 6270—The "Thermobank" system is based on a modified form of the reverse cycle. It is essentially a combination of a hot gas defrosting system plus a heat accumulator in the suction line and is designed to provide automatic defrosting for low temperature refrigeration systems. A 40 page bulletin (No. 16) published by the Kramer Trenton Co., Trenton 5, N. J., gives a comprehensive coverage of this system. Selection tables and considerable application data pertaining to load calculations are included.

More Efficient Heating With

AIRTHERM Direct-Fired Warm Air Heater

(Oil or Gas Fired)



Direct heat transfer through a single thickness of metal results in the high efficiency and low operating cost of the Airtherm Direct-Fired Warm Air Heater. The Airtherm is a self-contained heating system — ready to connect to your fuel and power supply lines.

Capacities from 650,000 to 1,950,000 B.T.U.'s. Either oil or gas-fired.

✓ *Write for Bulletin 801 describing this economical heater in detail.*

AIRTHERM

MANUFACTURING COMPANY

AT-43

711 So. Spring Ave. • St. Louis 10, Mo.

WILLIAMS

SUPERIOR DROP-FORGED TOOLS

"VULCAN" Drop-forged CHAIN PIPE VISES

Above: "VULCAN". 4 sizes for pipe 1/8 to 8 inches.

At right: "VULCAN SUPERIOR", with reversible jaws and adjusting handle on top. 2 sizes for pipe 1/8 to 4-1/2 inches.

"Vulcan", the original chain pipe vise, is made entirely of tough, wrought steel ... no cast parts whatever. The dropforged jaws are of saw-tempered steel for file sharpening ... chains are of the same high quality as Williams' "Vulcan" Tongs. Fully guaranteed, and sold by Industrial, Plumbing and Oil Field Distributors everywhere.

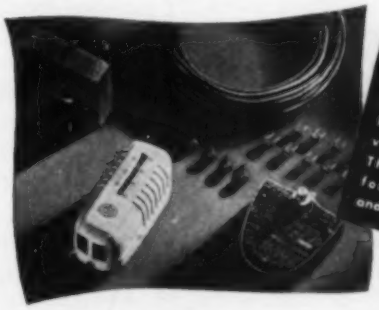
J. H. WILLIAMS & CO., BUFFALO 7, N. Y.

SELL A-P THERMOSTATIC HEAT REGULATOR SETS....To Owners of OIL HEATERS Sold Since 1939

You have a long list of ready prospects for this A-P Heat Regulator Set—all the owners of Oil Burning Space Heaters using any A-P Constant Level Oil Control sold and delivered since 1939.

Tell all these old friends how THERMOSTATIC TEMPERATURE REGULATION can be added to their heaters easily and at low cost — with the A-P Model 240-ED Heat Regulator Set. They'll appreciate the convenience and luxury of steady room comfort at the touch of a thermostat dial, and the fuel-saving advantage that will practically pay for the set.

Get ready for this EXTRA sale and profit now! Write for descriptive literature, prices, and samples of advertising and selling material.



A-P Model 240-ED Electric Heat Regulator Conversion Set Includes: Electric Conversion Top, Wall Thermostat, Transformer, Accessories and Instructions.

Easily Installed — Just mount Electric Conversion Top on present Manual Control and connect to Thermostat and Transformer.

(Be sure your *new* Heater Lines are equipped with A P DEPENDABLE OIL Controls — for faster sales, greater customer satisfaction.)

AUTOMATIC PRODUCTS COMPANY

2460 NORTH THIRTY-SECOND STREET, MILWAUKEE 10, WISCONSIN

A-P DEPENDABLE Oil Controls

DESIGNED TO ELIMINATE SERVICING

Burners for Heavy Bunker Oils

No. 6271—Burners designed for use with heavy bunker oils and in boilers of 12 to 650 hp are described and illustrated in a new 16 page catalog (No. 46) issued by the Enterprise Engine & Foundry Co., Combustion Equipment Div., 18th and Florida Sts., San Francisco 10, Calif. Features of the units include the v belt drive and a patented tailpiece for admitting fuel to the hollow burner shaft. Combination oil and gas burners are also covered.

Condensate Return System

No. 6272—The "C-B" system for returning condensate to boilers at high pressure and high temperature is described in publication No. 3250 issued by the Cochrane Corp., 3131 N. 17th St., Philadelphia 32, Pa. In this system, a specially designed centrifugal pump is used to energize an eductor, which in turn pumps the condensate from equipment drainage return lines into the boiler in a closed circuit.

Continuous Blow-Off Equipment for Boilers

No. 6273—Heat savings, quick return on investment, and maintenance of uniform concentrations are some of the advantages claimed for continuous blow-off systems as compared to intermittently blowing down a boiler it is stated in a new 12 page bulletin (No. 4081) issued by the Cochrane Corp., 3131 N. 17th St., Philadelphia 32, Pa. The company's line of continuous blow-off equipment is also described and illustrated.

For *permanent* insulating efficiency, leading refrigeration engineers have long relied on Rock Cork's many service advantages, such as its low thermal conductivity... resistance to moisture, vermin, rot ...and its imperishable, basically mineral, composition. Rock Cork is made in sheet and pipe covering form, is adaptable to every type of refrigeration construction.

Expert Application: J-M Insulation Applicators are organized to handle every detail of your insulating job. These men determine the most economical thickness of Rock Cork to use. And, their skillful application methods assure your obtaining the peak savings which only proper workmanship can make certain. Backing these construction units are J-M's years of experience in this field.

For complete information about Rock Cork, write for brochure DS-555... or get in touch with your J-M Insulation Applicator. Johns-Manville, Box 290, New York 16, N.Y.

JM PRODUCT

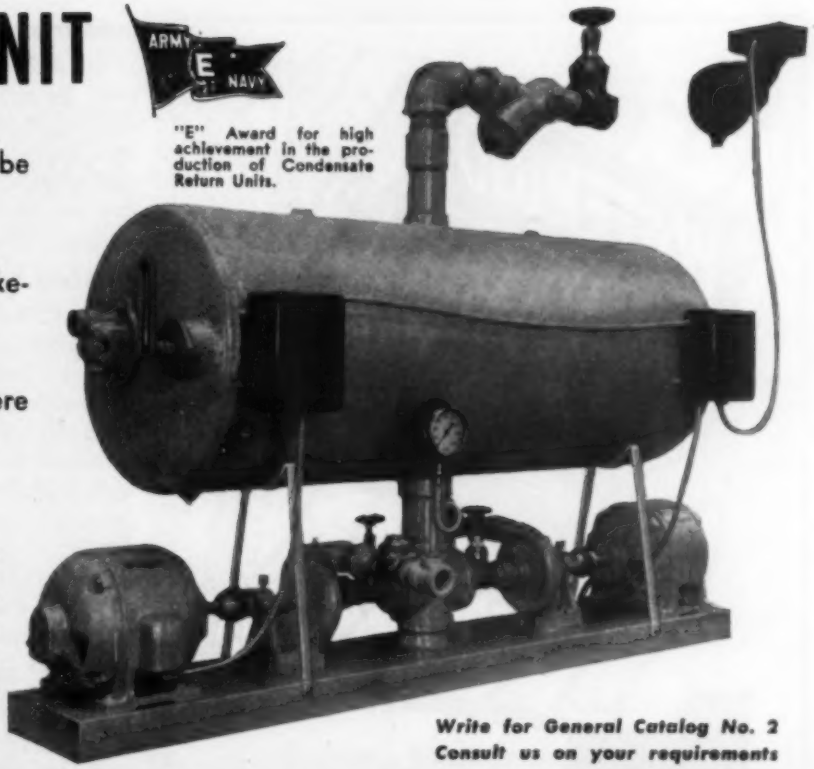
Johns-Manville
ROCK CORK

Auer Grilles
Auer makes many grille designs in stamped metals for all purposes. Furnished in sizes, materials, and finishes to fit your requirements for air conditioning, ventilating, radiator enclosure, or concealment. To simplify your specification and detail work on grilles, use the Auer Grille Catalog "G". It gives complete grille dimension tables with actual size details. A copy will come to you on request.
THE AUER REGISTER CO.
3608 Payne Ave.,
Cleveland 14, Ohio
THE AUER REGISTER CO. CLEVELAND, OHIO

BOILER FEED UNIT

"E" Award for high achievement in the production of Condensate Return Units.

- For use where boiler feed should be controlled by boiler water level.
- Where operation of boiler water make-up condensate return is combined.
- In process steam installations where little condensate is returned.



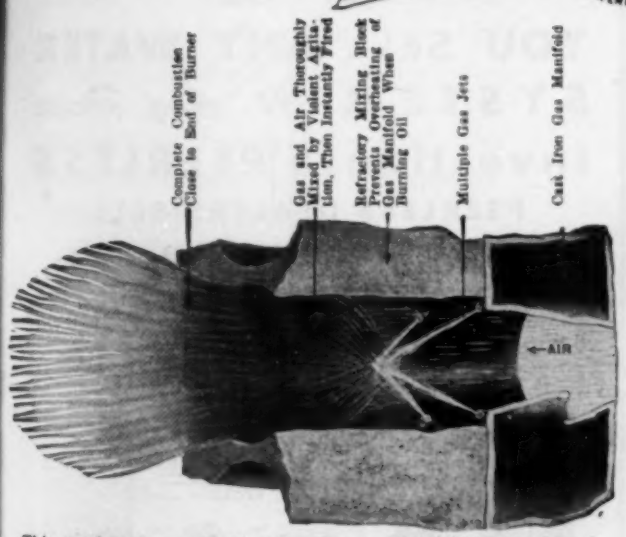
This is one of the various types of Condensate Return Units manufactured by Walter H. Eagan Co. No matter what your requirements, there is an Eagan Unit to do the job.

Write for General Catalog No. 2
Consult us on your requirements

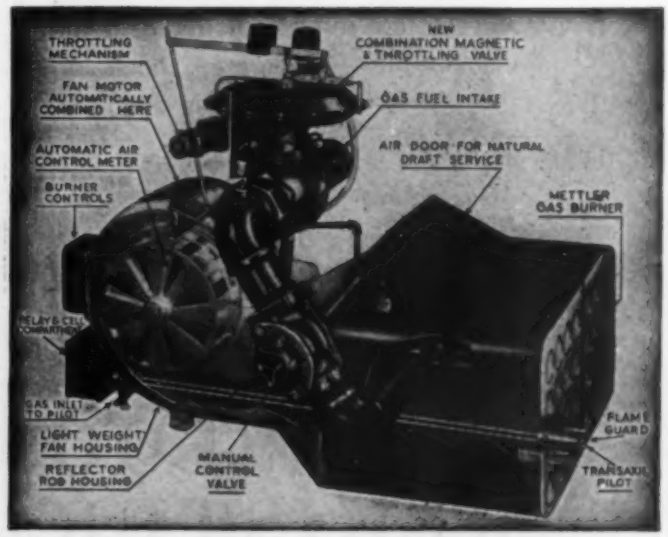
WALTER H. EAGAN CO.

2336-38 FAIRMOUNT AVENUE, PHILADELPHIA, PA.

The Mettler ENTRAINED COMBUSTION Gas Burners


Complete Combustion Close to End of Burner
Gas and Air Thoroughly Mixed by Violent Agitation, Then Instantly Fired
Refractory Mixing Block Prevents Overheating of Gas Manifold When Burning Oil
Multiple Gas Jets
Cast Iron Gas Manifold
AIR

This section thru a Mettler Gas Burner Manifold shows how it "mixes" gas and air. It's *easy* to see why it's still the best.

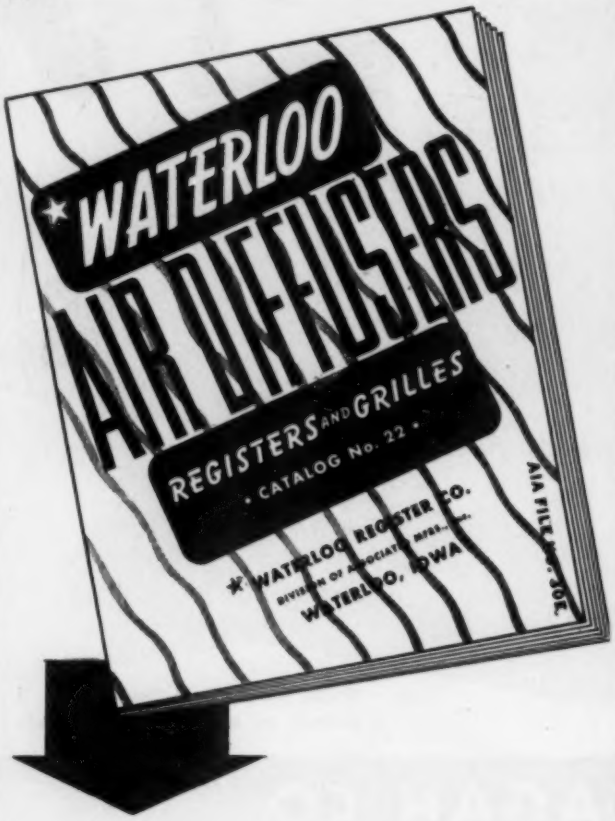


The "Fan-Air Gas Burner" in adjoining picture embodies this leading patented casting design with all the other essential and desirable features of this *Fully Assembled Mechanical Draft Automatic Gas Burner.* It comes to you complete as pictured, with simple, dependable, automatic controls. No other burner *so easy to install.* Why be satisfied with less—when all these tangible advantages are so readily available.

SIZES 5 TO 400 HP. **PROMPT SHIPMENTS**

LEE B. METTLER CO.

LOS ANGELES 13 — 406 South Main Street — CALIFORNIA



Your COPY IS WAITING!

Send today for this complete catalog and price list—it's up-to-the-minute and complete—gives every detail and fully describes Waterloo Registers and Grilles.

Also includes simplified grille selection data—so you can quickly determine grille sizes for supply and return openings. Now don't forget—but write *right away* for your free copy.

WATERLOO REGISTER SALES CO.

WATERLOO, IOWA

Industrial Oil Burners

No. 6274—A versatile line of oil burners, designed to burn No. 1 to No. 6 oils in boilers from 45 to 400 hp, is covered in a 12 page folder published by York-Shipley, Inc., Industrial Div., York, Pa. A patented flame-former, designed to distribute the flame evenly throughout the fire box and an automatic gas torch lighter, are some of the features of this horizontal rotary type burner.

Packless Expansion Joints

No. 6275—The Magnilastic Div. of Cook Electric Co., 2700 Southport Ave., Chicago 14, Ill., has issued a new catalog (No. 242M) illustrating and describing packless expansion joints, anchor joints, and low pressure large diameter expansion joints. Recent developments such as offset, turbo, and jacket joints are included, together with high temperature exhaust bellows and heavy duty clinch-lock bellows. The various types described cover a wide range of applications involving the transmission of steam, oil, gas, and many chemicals. Equipment is designed for use with pressures from vacuum to 1000 psi, temperatures from —300 F to 1600 F, and standard pipe sizes from ½ to 24 in.

Pilot Lights

No. 6276—Series PLN-849 pilot lights, designed for use with General Electric Co. NE-51 neon lamps, are described and illustrated in a 4 page booklet issued by the Dial Light Co. of America, Inc., 900 Broadway, New York 3, N. Y. These pilot light units feature built-in resistors and are available in a variety of lens colors and in a number of types for various applications.

BEFORE YOU SELL ANY WATER SYSTEM *At Any Price*

Investigate PEERLESS

PEERLESS DEALERS SELL THE 2 MOST ADVANCED *Domestic Water Systems*

PEERLESS WATER KING FOR SHALLOW WELLS

CAPACITIES 275 TO 840 Gallons per hour

PEERLESS JET SYSTEM FOR DEEP OR SHALLOW WELLS

PEERLESS PUMPS

Write today for full dealer information

PEERLESS PUMP DIVISION

Food Machinery Corporation

Canton 6, Ohio Quincy, Ill. Los Angeles 31, Calif.

PRODUCTION JUMPS...

WHEN YOU EQUIP
YOUR RETURN LINES WITH
Schaub DRAINTROLS

• Draintrols provide quicker heat-up at starting time . . . there are no delays waiting for equipment to reach operating temperature . . . AIR and CONDENSATE are discharged instantly . . . Machines hold uniform working heat all day long . . . *Net results are more production and greater profits* . . . Draintrols have no moving parts to get out of order —they replace ordinary traps on existing "open" systems.

Write today for full information.

FRED H. SCHAUB ENGINEERING CO., INC.
2116 S. Marshall Blvd. Chicago 23, Illinois

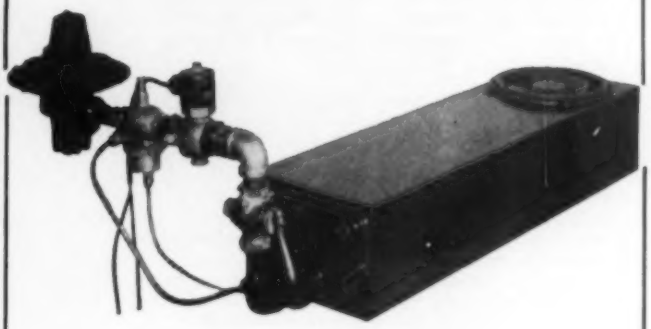
The SCHAUB Heat Reclaim System

Step up the steam capacity of your boiler with the same fuel, if you have a regular open system. Our patented hot water heat exchanger, with the Schaub Heat Reclaim System, transfers "flash" heat (ordinarily lost out of the vent) from the returns to the feed water on its way to the boiler. Resulting higher feed water temperature (240 degrees to 280 degrees) increases boiler capacity. This means little or no drop in steam pressure when feed water is admitted to the boiler.
Write for details.

The Key to Boiler Room Efficiency
SCHAUB SYSTEMS

JOHN ZINK BURNERS

ONE-STOP
BURNER
SERVICE



One of the Major Industrial Burner Manufacturers of the world offers to design and manufacture a special adaptation of this John Zink Conversion Burner for your particular boilers and furnaces enabling you to furnish complete gas-fired units.

JOHN ZINK PRODUCTS
ARE PROTECTED BY
U. S. LETTERS OF PATENT

"There's A John Zink Burner for Every Heating & Power Need"

John Zink Company
4401 South Peoria
TULSA, OKLAHOMA
New York - Detroit - Salt Lake City - Houston - Los Angeles

Wherever air freshness must be maintained for:

Control of Product Quality

Personnel Efficiency

Customer Satisfaction

Nuisance Abatement

. . . Dorex Air Recovery Equipment, employing highly activated carbon to remove gaseous and odorous contaminants, affords a positive and economical means of controlling the quality of air.

Dorex Air Recovery Equipment is adaptable to large heating, ventilating and cooling systems or to such compact units as are used in passenger vehicles.

For instance, over 90% of the new railroad coaches and Pullmans will use Dorex Air Recovery Units. There is so little space for air conditioning equipment in railroad cars that older systems must sacrifice temperature control if more than 25% outdoor air is added—and 25% often is too little for comfort. But, fitted with Dorex carbon adsorbers, the same apparatus gives the equivalent of 75% air renewal—usually more than sufficient for passenger comfort.

We will be glad to send the details of other typical installations. For full information and quotations, call the nearest district representative or write Dept. D4, 112 East 32nd Street, New York 16, New York.

W. B. CONNOR ENGINEERING CORP.

AIR DIFFUSION AIR PURIFICATION AIR RECOVERY

114 East 32nd Street CONNOR New York 16, N. Y.

IN CANADA: Douglas Engineering Co., Ltd., 1405 Bishop St., Montreal 25, P. Q.

Units may be directly connected to 110 and 220 volt circuits. Resistor value is chosen to suit the voltage and may be varied to obtain more light or longer life for special conditions.

Portable Pyrometers

No. 6277—Bulletin No. D602-4, issued by Wheelco Instruments Co., 847 W. Harrison St., Chicago 7, Ill., describes the company's line of portable pyrometers. Specifications for the proper selection of extension holders, adapters, and thermocouple tips are listed in chart form and information pertaining to a new extension type pyrometer and the Model 2863 portable pyrometer kit is included.

Process Controllers

No. 6278—Instruments and controls for process industries are covered in a new bulletin (No. 17) published by Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, Ohio. Process controllers, control components, measuring components, and instrument combinations are briefly described and illustrated. Application data such as sizes, pressure standards and ranges are listed and typical control systems for both the single and multiple element type are illustrated.

Protective Coating Comparison Chart

No. 6279—A comprehensive 8 page bulletin gives information in condensed form on the entire line of plastic protective coatings offered by the Amercoat Div., American Pipe & Construction Co., P. O. Box 3428, Terminal Annex, Los Angeles 54, Calif. Charts show the characteristics and properties of the various coatings and

PREHEAT FOR PROPER HEAT

WHITLOCK
FUEL OIL HEATERS

The Whitlock Type V-1 Oil Heater shown above is of simple standard design at moderate cost. These units in all-welded construction, adapted to high working pressures are *immediately available from stock.*

Whitlock Fuel Oil Heaters of more elaborate design are made to suit individual operating conditions and engineered for any combinations of temperature rise, operating pressure and capacity. Incorporating Whitlock's experience of many years in the fabricating of such equipment, they guarantee a unit to suit your requirements for economy and efficiency.

Our engineers will gladly discuss your fuel oil heating problems and make specific recommendations. Write for new Bulletin No. 35 describing and illustrating the complete line of Whitlock Fuel Oil Heaters.

THE WHITLOCK MANUFACTURING COMPANY

44 South St., Elmwood, Hartford 1, Conn.

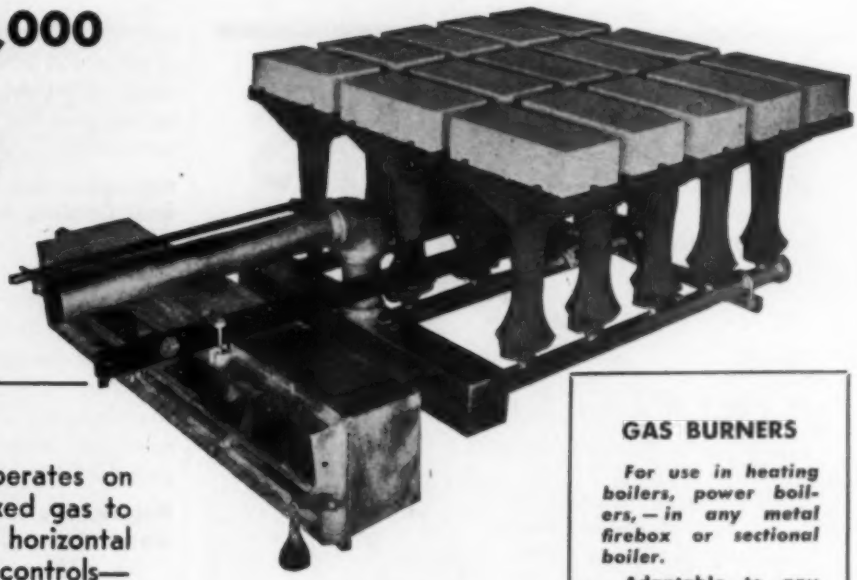
New York Boston Chicago Philadelphia Detroit Richmond

Authorized representatives in other principal cities.

In Canada: Darling Brothers, Ltd., Montreal

Established 1891 — Ahead of the Times for Half a Century

50,000 to 10,000,000 B. t. u.
WECO—N. G. E.
Series F600
GAS BURNERS



The Series F600 Burner operates on straight natural gas and mixed gas to 800 B.t.u. Gives perfect horizontal distribution. Full automatic controls—that operate properly—can be easily applied.

Write for
F 600 Bulletin

GAS BURNERS

For use in heating boilers, power boilers, — in any metal firebox or sectional boiler.

Adaptable to any firebox shape or size.

Handles various gas pressures and affects low draft loss.

THE WEBSTER ENGINEERING CO.

TULSA —DIVISION OF— **OKLAHOMA**
Surface Combustion . . . . . Toledo, Ohio

THE NEW AERO-PASS CONDENSER
...with Automatic Control for both summer and winter operation

TO INCREASE YOUR PLANT OUTPUT OF PRODUCT, FIRST OPEN THE BOTTLE-NECK OF REFRIGERATION CAPACITY.

Install the NIAGARA AERO-PASS CONDENSER and your refrigeration plant operates with a smaller differential between suction pressure and head pressure, gaining for you both an increase in capacity from your compressors and important savings in expense for power.

The New NIAGARA AERO-PASS CONDENSER offers its advantages protected from the danger of loss through shut-downs and maintenance troubles. The 10-year record of Niagara Condensers for dependable full capacity operation is unequalled.

Write, today, for the details of construction, operation and power and water savings. Ask for Condenser Bulletin 101-HP.

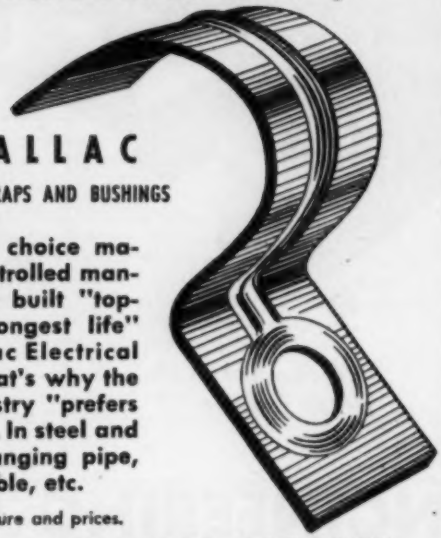
NIAGARA BLOWER COMPANY
Over 30 Years of Service in Industrial Air Engineering
405 Lexington Ave. New York 17, N. Y.
Field Engineering Offices in Principal Cities

NIAGARA * **AERO-PASS CONDENSER**
*Trade-mark Registered for the Niagara Condenser with the patented Duo-Pass, Oilout and Balanced Wet Bulb Control.

Millions of Steel

JIFFY CLIPS

Serving the Electrical Industry



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MINERALLAC

HANGERS, CLIPS, STRAPS AND BUSHINGS

Expert design, choice materials and controlled manufacture have built "top-service and longest life" into Minerallac Electrical Specialties. That's why the electrical industry "prefers Minerallac"... In steel and Everdur for hanging pipe, conduit, BX cable, etc.

Send for new literature and prices.

MINERALLAC ELECTRIC COMPANY
25 NORTH PEORIA STREET—CHICAGO 7, ILLINOIS

MINERALLAC

provide information for selecting the proper coating, preparation of the surface, and application methods on steel, concrete, and wood. In addition, the cost per square foot for various materials with area formulas is provided.

Refrigeration Coils and Cooling Units

No. 6280—Tenney Engineering, Inc., 26 Avenue B, Newark 5, N. J., has issued a new 20 page catalog (No. CD46) on cooling units. Among the products included are unit coolers, "Kwik-Freeze" ice makers, coil and pan combinations, wall mounted panel units, and various types of coils such as bare tube, bunker type, display case, gravity, and others. Included also are sizes, prices, selection tables, formulas for determining coil sizes, and useful application data.

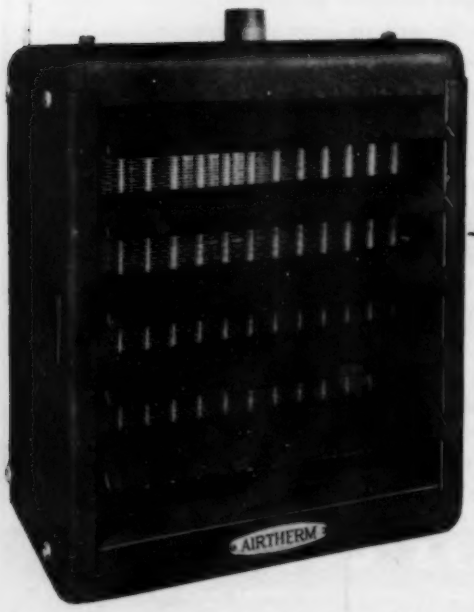
Regulators for Combustion Controls

No. 6281—The Ace Instrument Co., 123 W. 21st St., New York 11, N. Y., has issued a new 12 page booklet covering the company's line of electrically operated combustion control regulators. The line of regulators includes units for maintaining proper steam pressure, the correct amount of air, and the proper furnace draft. Various arrangements are available for use with stokers, pulverized fuel, gas, and oil burners.

Silicone Insulating Varnish

No. 6282—"DC-996" silicone varnish, which is said to cure at ordinary baking temperatures and to protect electrical equipment operating continuously at temperatures up to 350 F and in the presence of excessive mois-

Individually Controlled Airtherm Propeller Fan Unit Heaters Will Cut Your Fuel Bill

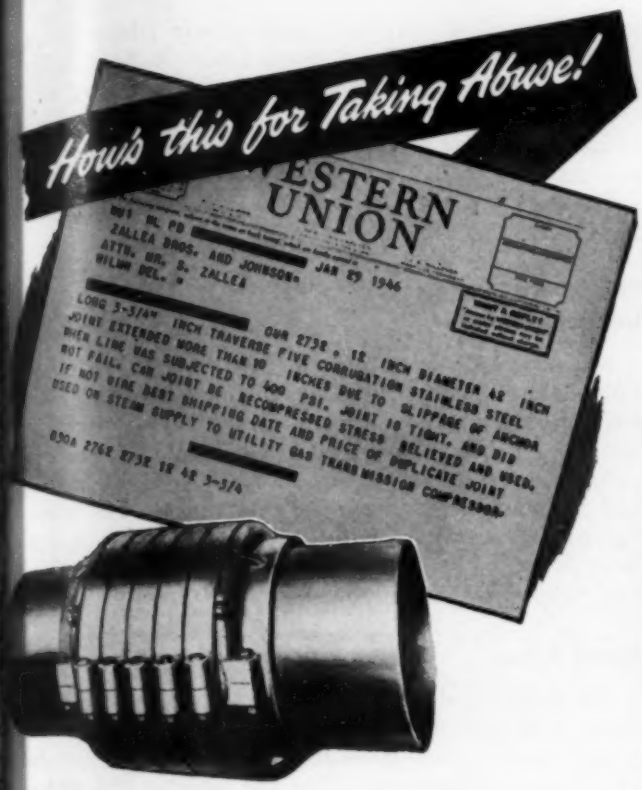


AIRTHERM Propeller Fan Unit Heaters will reduce your fuel costs.

Heat can be uniformly distributed through the use of AIRTHERM Heaters with individual controls, eliminating dead spots in your plant and increasing your heating efficiency.

There is an AIRTHERM Heater for every heating application. Write for Bulletin 1206 giving complete range of available sizes.

AIRTHERM
MANUFACTURING COMPANY
711 South Spring Avenue • St. Louis 10, Missouri



THE Zallea Self-Equalizing Expansion Joint referred to in the above wire was installed in a 250 lb steam line. On test, 400 psi was applied, and, because of an error in calculation of required anchor strength, the main anchor on the blank end of the line "let go." Before the condition could be corrected the joint extended more than 10"—nearly 3 times its rated maximum traverse—*but it did not fail.*

No expansion joint was ever designed to withstand the abuse to which this one was subjected. But the quality we build into every type we produce makes them endure—even in the severest service.

Zallea Corrugated Expansion Joints are made in non-equalizing, self-equalizing and duo-equalizing types, depending upon the conditions under which they must operate . . . of copper, stainless steels and other corrosion-resistant alloys . . . diameters from 3" upward . . . traverses up to 7½" . . . for pressures from vacuum to 300 psi . . . for temperatures from sub-zero to 1600°F. Let us quote on your requirements.

"Code-Craft"...

ZALLEA BROTHERS & JOHNSON

EQUIPMENT FOR THE PROCESS INDUSTRIES

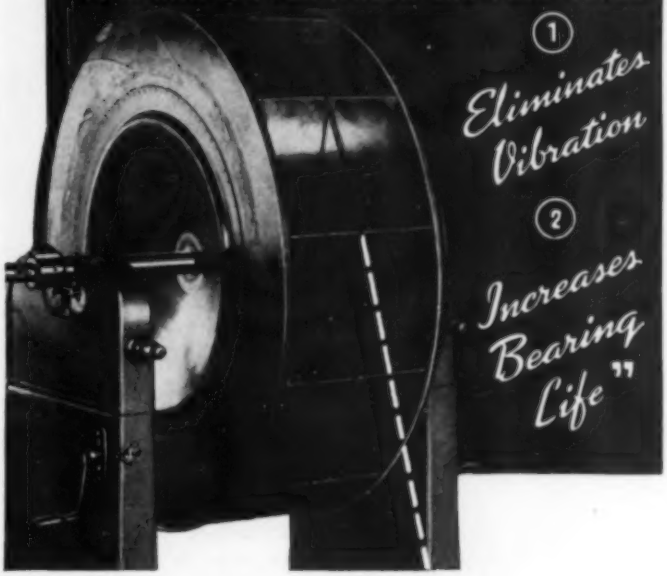
816 LOCUST ST. • WILMINGTON 99, DEL.

Specify and demand ZALLEA Furnace-Annealed Expansion Joints

Reports ILG Electric Ventilating Company:

"BEAR BALANCING

of Centrifugal Fan Wheels and Vane-Axials



Your Customers...

"Notice the Difference"

...when parts are DY-NAMICALLY BALANCED

—Because the machines you build and sell perform better . . . run quieter . . . vibrate less . . . wear longer . . . require less attention—when rotating parts are Statically and Dy-Namically Balanced.

USE FAST, EASY, LOW-COST BEAR BALANCING

There's a Bear-engineered Static and Dy-Namic Balancer for each rotating part, from tiny armatures to giant turbines. For instance, Heavy Duty Model 400, shown here, is designed to handle rotating bodies ranging up to 4,000 pounds.

Whichever Bear Balancer you choose, you will get simpler operation (*any shop-hand can learn Bear-Balancing in half a day*); quicker buyer-acceptance (*Bear machines balance rotating bodies to all accepted standards*); reduced costs (*Bear-Balancer-Speed cuts per unit manufacturing costs; Bear reasonable prices cut capital investment required*). To meet and beat growing competition, send for Bear-Balancing Data Bulletin. Write today to BEAR MFG. CO. Industrial Division, Dep't HPAC, Rock Island, Illinois.

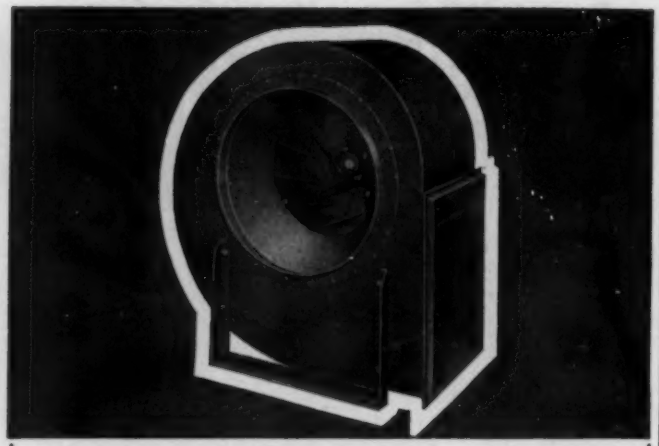
Drive Out the "Wobblies" of Static and Dy-Namic Un-Balance.

193

BEAR

COMBINATION

STATIC AND DY-NAMIC BALANCING MACHINES



'GENERAL' MULTIBLADE EXHAUSTERS

Exactly adapted to the Exacting requirements of Heating, Ventilating and Air Conditioning Services

Forward and Backward Curved Blades in all sizes, wheels 12 to 72 in.

And forward curve "Juniors" with wheel diameters 6 to 10

For your files, on request: Bulletin SC-102, with complete performance tables, dimensions, etc.—52 big pages.

GENERAL BLOWER CO.

"Lungs for Industry"

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District Sales: Chicago 10, Ill. 502 N. Dearborn St.

Sales Representatives in Principal Cities

QUALITY Always! MARTOCELLO Products are fully DEPENDABLE

ATOMIZING SPRAY NOZZLES, SPRAY POND NOZZLES, and CONDENSER WATER DISTRIBUTORS

For Vertical Shell and Tube Ammonia Condensers

ALL MARTOCELLO SPRAY NOZZLES are manufactured with precision and of a design which has been thoroughly tested for maximum results and durability. They are guaranteed to give satisfaction. Successful, efficient results depend largely upon selecting the proper number, type and size of Nozzles suitable for your installation.

THE MARTOCELLO CONDENSER WATER DISTRIBUTORS are now used as standard equipment by Progressive Refrigerating Engineers in solving their Manpower problem, because they require no attention and assure users of the lowest Condenser Operating Pressures and Minimum Power Cost.

It will pay you to consult us. WRITE or WIRE for further information.

Jos. A. Martocello & Co. 229-231 NORTH 13th STREET PHILADELPHIA 7, PA.

ture, is described in a new 4 page bulletin (Resin Series No. 1) issued by the Dow Corning Corp., Midland, Mich. This pamphlet is published primarily for men who operate rewind and electrical maintenance shops and contains practical instructions on how to apply and cure this material. Also included are tables giving its properties and specifications and a summary of its advantages.

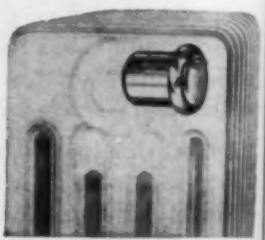
Slip Ring, Polyphase Induction Motors

No. 6283—Century Electric Co., 1806 Pine St., St. Louis, Mo., has issued a new eight page bulletin (Form 643) on slip ring, polyphase induction motors from 1 to 350 hp. Features are said to include high starting torque, low starting current, adjustable speed control, and adaptability to reversing operations. The manufacturer recommends these motors for many applications including reciprocating pumps, compressors, blowers, stokers, cranes, and hoists.

Small Relays

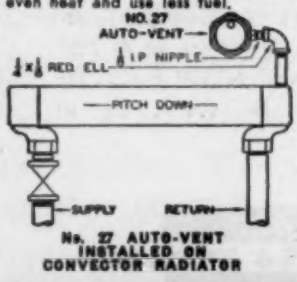
No. 6284—Coil and contact data, contact arrangement diagrams, dimension sketches of front and rear mounted units, and enclosing cover data are included in a 4 page booklet describing the Bulletin 104, midget metal base relays offered by the Ward Leonard Electric Co., 31 South St., Mt. Vernon, N. Y. These relays are recommended by the manufacturer for us in control circuits and applications where space is limited.

NO MORE AIR-BOUND RADIATORS



WITH THE *NEW* AUTO-VENT AIR ELIMINATOR

Install the No. 27 Auto-Vent Air Eliminator on all radiators and convectors and release that invisible Air Brake that retards circulation. As air accumulates in the radiators or pipe lines it is automatically vented, assuring perfect circulation throughout the system at all times. You get more even heat and use less fuel.



No. 27 AUTO-VENT INSTALLED ON CONVECTOR RADIATOR

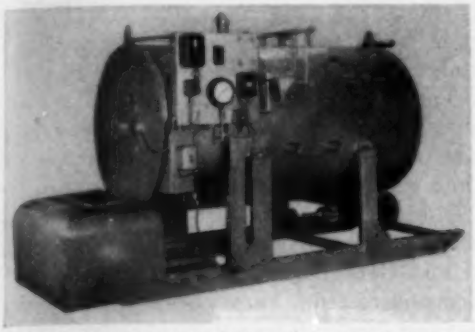
No. 27 can be installed on convectors, baseboard radiation panels, radiant heating coils, etc., without the use of an air chamber. For pressures up to 50 lbs.

Ask your jobber— Prompt delivery.

PLAY SAFE—USE THEM ON EVERY JOB

MAID-O-MIST INC. AUTOMATIC HUMIDIFIERS

3215 N. PULASKI RD., CHICAGO 41, ILLINOIS



Model SPL-30-5 Steam-Pak—30 h.p. Low Pressure

When you own a

STEAM-PAK GENERATOR

you own the finest steam generating system ever produced in the United States

Catalog ID-46-7D gives you complete specifications on the *Steam-Pak* line. It's yours upon request. Write today.

YORK SHIPLEY INCORPORATED

America's Most Complete Line Industrial, Commercial, Domestic Oil-Fired Equipment

Industrial Division YORK • PENNSYLVANIA

EFFICIENT ECONOMICAL

De Luxe type WK Pressure Actuated Water Regulating Valves for Freon, Sulphur or Methyl, are pilot operated, modulating. Standard adjustable opening range from 50 to 150 lbs. Factory set to open at 90 pounds. Precision engineering and individual testing for low cost, satisfactory performance. Available in sizes of ¾ to 2 inches FPT. Simple adjustment—easily serviced.

Other sizes and types are also available.

Electrimatic

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Consoco ammonia heat exchanger installation, Stanley Theatre, Philadelphia.

CONSECO

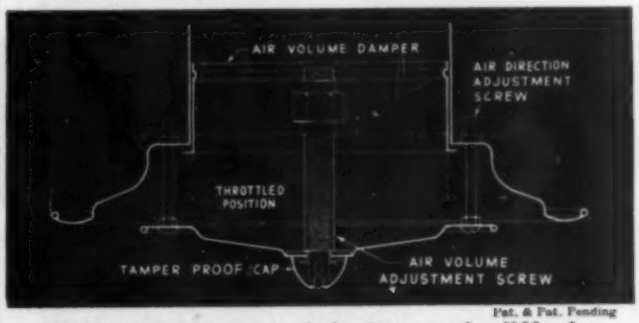
Twenty years' specialized experience designing, fabricating and servicing heat exchangers assure sound design, quality construction and economical operation with any type refrigerant. Engineering assistance gladly furnished. Write.

HEAT EXCHANGERS for Air Conditioning

SERVICE

CONDENSER SERVICE & ENGINEERING CO., INC.

81 RIVER STREET, HOBOKEN, N. J. • *Phone Any Time*



Pat. & Pat. Pending

All the advantages of draftless air diffusion . . . *plus* FAST SYSTEM BALANCING AIR PATTERN CONTROL

because Kno-Draft Air Diffusers are adjustable

Volume Damper for fast system balancing regulates the air outlet aperture uniformly without affecting the outlet velocity or diffusion pattern.

Air Direction Adjustment affords any angle of air discharge from vertical to horizontal to accommodate ceiling height, individual, seasonal, or air pattern requirements.

Specify Kno-Draft Diffusers for better mixing control of room and supply air, more uniform temperatures throughout the occupied zone and noiseless, draftless air distribution.

Send for FREE handbook containing sketches, charts, dimension prints and instructive text that simplify the selection and installation of air diffusers. For your copy, please write Dept. D2 using your company letterhead.

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AIR DIFFUSION AIR PURIFICATION AIR RECOVERY

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IN CANADA: Douglas Engineering Co., Ltd., 1405 Bishop St., Montreal 25, P. Q.

Solve it . . . with SOL-VET "08" WATER TREATMENT FORMULA

• **SOL-VET "08"** is a fast, efficient cleaner of scale, rust, slime, dirt, algae for all equipment using water for cooling. By dissolving impurities completely, they are carried out *in solution* and clogging is eliminated. Head pressures fall to nosmal; heat transfer is raised to maximum; full circulation is restored; life of equipment prolonged. On large jobs, shutdown time is cut to a maximum of 6 to 8 hours. **GUARANTEED EFFECTIVE.** If you find SOL-VET "08" not exactly as represented, when used according to instruction and full, accurate report is given within 30 days from date of purchase, your money will be refunded. **GUARANTEED HARMLESS** when used as directed, to all water conducting materials in water cooled equipment.

ANDERSON-STOLZ CORPORATION
1731-33 WALNUT ST. KANSAS CITY 8, MO.

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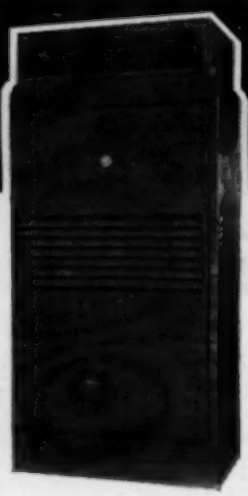
A RESPECTED NAME IN AIR CONDITIONING



3 and 5 TON SELF-CONTAINED UNITS

All-purpose, handsomely designed, smooth functioning air conditioning units with all copper condenser and liberally designed coils—removable outlet box for use with duct system—removable panel in back of outlet box for additional grills where front and back air distribution is desired in center of room . . . two way adjustable deflecting grill . . . knock-out for fresh air connection . . . removable one-piece panel in front for easy accessibility. The cabinets are made of furniture steel . . . crackle finish. **Also available with one or two row heating coil.**

TYPHOON PRODUCTS

PROP-R-TEMP *Self-Contained Air Conditioning Units, 3 and 5 Tons.*
TYPHOON LOW-SIDES, *3 and 5 Ton Units*
TYPHOON EVAPORATIVE CONDENSERS.
MONSOON *Heating, Direct Expansion and Water Coils.*

TYPHOON PRODUCTS

ICE AIR CONDITIONING CO., INC.
794-6 UNION STREET · BROOKLYN 15, N. Y.

Steam Generator and Oil Burner

No. 6285—An eight page folder, issued by York-Shipley, Inc., Industrial Div., York, Pa., describes the company's line of "Steam-Pak" generators and its line of horizontal, rotary oil burners. The generators are designed to burn oils from Nos. 3 to 6 and are offered in two series. The low pressure series is available for maximum working pressures of 15 psi steam or 30 psi water, and units are rated from 2100 to 13,500 EDR steam. The high pressure series is available for maximum working pressures of 100 psi and in 16 sizes from 4 to 100 hp. Generators are self-contained and factory assembled and tested. Features of the oil burners are described and illustrated and specifications for the nine sizes are included.

Steel Applications

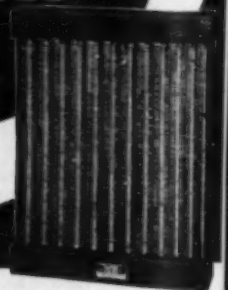
No. 6286—Issue No. 18 of an 8 page newspaper size rotogravure paper has been published by Joseph T. Ryerson and Son, Inc., Box 8000-A, Chicago, Ill., distributors of steel from warehouse stocks. Consisting almost entirely of steel applications pictures, the paper illustrates the diversified uses of steel throughout the metal working and fabricating industries and also in other businesses from circuses to salt mines.

Two Stage Centrifugal Pumps

No. 6287—Ingersoll-Rand Co., 11 Broadway, New York 4, N. Y., has published a 16 page catalog (Form 7062) covering the company's Class GT, two stage centrifugal pumps which are suitable for many industrial

Continuous and Reliable HEAT!

CHROMALOX
Electric
AIR-BLAST HEATERS
IN AIR DUCTS FOR DRYING OVENS, SPACE HEATING, ETC.



·ECONOMICAL ·EASY TO INSTALL

CHROMALOX Air-Blast Units are a reliable heat-source in air ducts for heating air under forced circulation. Banks of Fin-strip Heaters deliver more heat in a given area. Fins dissipate heat faster, permitting lower operating temperatures.

Standard CHROMALOX Air-Blast Heaters are available in seven sizes with capacities from 5-KW to 40-KW.

FOR IDEAS . . .
Write for "100 Ways To Apply Electric Heat" booklet and data-packed Catalog 42.

EDWIN L. WIEGAND CO., 7610 THOMAS BLVD., PITTSBURGH 8, PA.

CHROMALOX
ELECTRIC HEAT FOR MODERN INDUSTRY

EVOLUTIONARY IN ENGINEERING
REVOLUTIONARY IN DESIGN

COOLSTREAM CAPACITY CLIMBS

Rising above original estimates—
Delivering more than promised—

Coolstream Electric Water Coolers
Exceed Expectations and
Surpass Specifications

The CP-10 is delivering more
than the charted 10 gallons—
The CP-10 is breaking
its own record!

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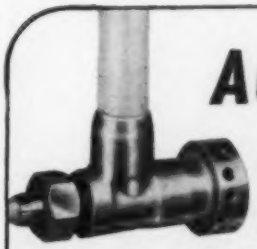
THE COOLSTREAM CORPORATION
240 BUTLER ST., BROOKLYN 17, N.Y.



Automatic Liquid Feed without Moving Parts

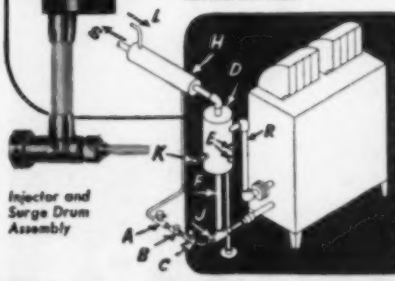
Injector with Distributor Head

THE PHILLIPS INJECTOR provides completely automatic evaporator feed with *no moving parts*. By supplying recirculated liquid steadily to the evaporator it increases heat transfer and increases the evaporator capacity.

Many existing dry-expansion evaporators can be easily changed to Injector operation, with improved performance assured.

Adaptable to freon applications, from air conditioning and water chilling to cold storage installations with plate coils, fin coils, and unit coolers. Injector capacities from fractional to 200 tons with freon. Furnished with either single outlet or distributor head.

Write for sketches showing typical Injector installations.



Injector and Surge Drum Assembly

FLOOR TYPE ROOM COOLER HOOK-UP
A—MAGNETIC SOLENOID
B—BULLS-EYE
C—HAND VALVE
D—SURGE DRUM
E—SIGHT GLASSES
F—LIQUID LEG
H—HEAT EXCHANGER
J—INJECTOR
K—CHARGING CONNECTION
L—LIQUID LINE
R—GAS RETURN
S—SUCTION LINE

H. A. PHILLIPS & CO.
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Shown above is the Hartzell Belt-drive Vaneaxial Blower. Available also in Direct-drive models. 18 to 42 inches.

MORE AIR PER DOLLAR AGAINST PRESSURE

How much can Hartzell save for you in air delivery costs? Reduce power consumption just 5 horsepower and over a year of 300 8-hour days at 1½ cents K.W.H. you save $165.00. Certified Test Code figures will open your eyes to the proved economy of air delivery against high pressures afforded by the Hartzell Vaneaxial Blower. Repeated tests show a peak total efficiency of 87%. Quiet and economical against pressures of 8" water gauge and more. Superior performance of one Hartzell Vaneaxial Blower prompted a leading manufacturer to buy 106 more! Get all the facts. Write for Bulletin 1601 today!

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PROPELLER FAN CO.
PIQUA DIV. OF CASTLE HILLS CORP. OHIO

YARWAY SPRAY NOZZLES

for • SPRAYING • WASHING
• RINSING • COOLING
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FAN-TYPE FOR FLAT SLICING SPRAY

Get the most out of your Spraying Equipment.

Use Yarway Nozzles. No internal vanes or other restrictions to clog or hinder flow. Two types—Yarway Involute-type producing a fine hollow spray with minimum energy loss, and Yarway Fan-type producing a flat fan-shaped spray with time-saving slicing action for cleaning.

Wide range of sizes and capacities. Thousands in use. Write for new Bulletin N-616.

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EASY TO READ
DIAL
THERMOMETER
Economize WITH POWERS
TEMPERATURE REGULATORS
POWERS
Use a Powers No. 11 Temperature Indicating Regulator when you want the advantages of an easy-to-read dial thermometer combined with a dependable self-operating regulator. The dial thermometer gives a visual check on the performance of the regulator and makes it easy to adjust for the required operating temperature. Various dials and ranges are available.
Is Easy To Install—because both the thermometer and the regulator operate from the same thermal system—only 1 tapped opening is required.
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THE POWERS REGULATOR CO.
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CONTROLS HEATING OR COOLING MEDIUMS
POWERS
OVER 50 YEARS OF AUTOMATIC TEMPERATURE AND HUMIDITY CONTROL

YOU GAIN BY GETTING
BOTH FROM BROWNELL

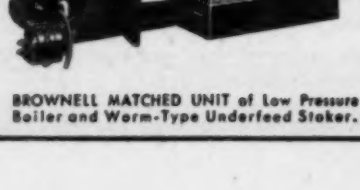
When you have a BROWNELL MATCHED UNIT, only one manufacturer is accountable for both the boiler and the stoker. There's no division of responsibility. More than that, you know that you have good heating equipment—with a great reputation back of it.

A BROWNELL MATCHED UNIT gives you all the benefits of mechanical firing . . . smokeless combustion; automatic regulation; fuel economy; saving of time, labor, and worry.

Brownell boilers and stokers offer you a wide range of selection to meet your requirements. Along with Brownell equipment goes competent engineering service.

The Brownell agent in your vicinity will be glad to give you full information. If you don't know his name and address, write us.

THE
BROWNELL COMPANY
408 N. FINDLAY ST.
DAYTON 1, OHIO



BROWNELL MATCHED UNIT of Low Pressure Boiler and Worm-Type Underfeed Stoker.

applications. These ball bearing pumps are designed to operate at modern motor, engine, or turbine speeds and are available in capacities up to 2200 gpm for discharge heads up to 450 psi.

Valves and Fittings

No. 6288—The Henry Valve Co., 3260 W. Grand Ave., Chicago 51, Ill., has issued a new catalog (No. 65) covering ammonia valves and fittings. In addition to shut-off valves, check, relief, purge, and three-way valves are included. Scale traps, gage sets, unions, and charging connections are also covered. Catalog No. 67 covers the company's line of drop forged and cold rolled steel fittings. Elbows, tees, plugs, couplings, and reducing bushings are illustrated.

Valve Specialties

No. 6289—Pressure relief, combined relief and check, check, stop and check, pressure and temperature relief, combined pressure, temperature and vacuum relief, together with reducing and regulating valves are featured in a new 20 page catalog issued by the Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill. Applications, sizes, pressure and temperature ranges, and construction features are included.

Ventilating Shutter

No. 6290—The "Air-Flo" automatic shutter, incorporating counterbalanced guides on the inside of the frame, is described in a new catalog (No. 41) published

"COMMERCIAL"
STANDARD
STEEL SHAPES

The latest edition of our shape book is now being mailed.

It supersedes all others...if you never received one before write for this new edition—
"Shape Book No. 7."

Listing circular steel shapes available without die or tool charges....A necessity for all buyers interested in circular stampings.

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YOUNGSTOWN, OHIO

Better AIR CONDITIONING *at Lower Cost*

GOVERNAIR

AIR CONDITIONING SYSTEMS

Here's what Governair Air Conditioning Systems offer you:

1. Complete, self-contained unit assemblies for greater operating efficiency.
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GOVERNAIR Self-contained Units are available in a wide range of sizes, enabling you to secure a system that is tailor-made to suit every special requirement.

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AT-A-GLANCE OIL GAUGE

PERFECT VISIBILITY!

EASY TO INSTALL!

ONE ADJUSTMENT ONLY!

An exclusive patented feature is the double enclosure of glass and plastic. Another exclusive patented feature is the calibration assembly that prevents oil from seeping into the calibration dome.

Easy to install in all basement tanks, even when partially full.

made at the factory to fit all tank sizes, and according to individual specifications.

THE ONLY NON-MAGNETIC GAUGE LISTED AS STANDARD BY UNDERWRITERS LABORATORIES

Write for complete details

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GREEN BAY, WISCONSIN

Positive Protection

T-IN at FACTORY

For all your under- and exposed piping you'll like DURANT Sealed Insulated Pipe. waterproof . . . re- installation costs to a um . . . eliminates olysis and corrosion requires very little ing and field work . . . extra cost of waste breakage . . . comes to ready to install. In D.I.P. is the *perfect* for this type of job. igate D.I.P.! Write tails today.

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ULATED PIPE CO.
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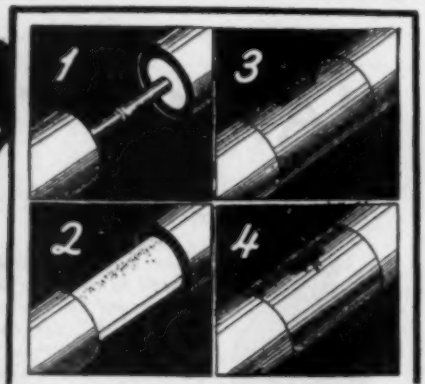
Palo Alto, Calif.



4 Simple Steps

1. Field joint ready for inspection. 2. Joint covered with standard pipe insulation. 3. Durant joint casing in place ready for asphalt. 4. Asphalt poured in slot making a perfect seal.

Ready to Install

FOR STEAM HOT WATER REFRIGERATION

D.I.P. TRADE MARK

The ORIGINAL Pre-sealed Insulated Pipe

Save Time, Trouble and Money

Seal Pipe Joints and Gaskets with Leakproof KEY Graphite Paste

Key Graphite Paste will hold a pipe joint leakproof against all petroleum based liquids. Key lubricates as it seals and positively will not freeze the joint.

Key Paste is listed by the Underwriters' Laboratories.

Your jobber will supply you or send for liberal sample and try Key at our expense.

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Please forward free sample of Key Graphite Paste.

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Just use it once—

The Alnor Velometer

gives you instantaneous, direct readings of air velocity in feet per minute. Hold it in the air stream for low range readings; use the tube-connected special jets for high range readings or inaccessible locations. That's all there is to it, and once you use the Alnor Velometer you will never tolerate other methods.

The Alnor Velometer is built in several standard ranges from 20 fpm to 6,000 fpm, and up to 3 inches static or total pressure. Special ranges available as low as 10 fpm, and up to 25,000 fpm and 20 inches pressure. Write for Velometer bulletin.

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Manufacturers of

Industrial Indicating Remote Reading and Recording Thermometers

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Controllers for Time, Pressure and Temperature

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Pressure Gauges and Recorders

GOTHAM
DUAL PERFORMANCE
THERMOMETER
4½" Dial
Used in lieu of a recorder
Red hand moves only when
rise of temperature occurs

GOTHAM
INSTRUMENT CO., Inc.
NEW YORK • CHICAGO • SAN FRANCISCO
Representatives in all Principal Cities
Illustrated catalog H on request

The best protection is PERFECTION

by the Air Conditioning Products Co., 2340 Lafayette Blvd., Detroit 16, Mich. Shutter vanes are designed to open 85 deg when the fan is on and to snap shut when the fan is not operating. Available sizes from 10 in. x 10 in. to 60 in. x 60 in. are listed.

Welded Boiler Tubes

No. 6291—Joseph T. Ryerson and Son, Inc., Box 8000-A, Chicago 80, Ill., steel distributor with warehouses in 11 cities, has recently been appointed a distributor of the electric resistance welded boiler tubes manufactured by the Babcock & Wilcox Tube Co. An 8 page bulletin has been published giving information on fabrication, testing, and specifications of these boiler tubes, together with a description of this distributor's quick delivery service.

Welding Alloys

No. 6292—A special issue of The Eutectic Welder, monthly house organ of the Eutectic Welding Alloys Corp., 40 Worth St., New York 13, N. Y., features many applications of the company's products to wartime metal joining problems. Two pages are devoted to pictures of the company's laboratories where new alloys and fluxes are developed. The manufacturer states that these low temperature welding products are well suited to peacetime production applications where problems of distortion, stresses, and strains are often caused by welding at higher temperatures.

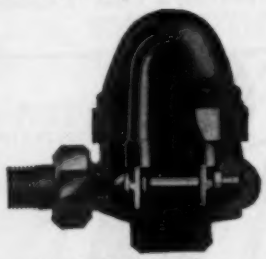
HAINES TRAPS

Designed for the job

F & T Trap

Haines Float and Thermostatic traps are designed to help you lick the toughest jobs. Use them for any low pressure job up to 20 lbs. in the ¾" size and 30 lbs. in the 1" and 1¼" sizes; they cannot become air bound, and the Haines underslung valve principle assures safety . . . no danger of building up a pressure.

The famous Bourdon Thermostatic tube is the operating member of Haines traps, and is the strongest ever built into a steam trap. The Bourdon Tube is used in gauges where accuracy is paramount, and you'll find that is the reason Haines traps always give reliable performance.



Vento Radiator Trap

For examinations under pressure simply remove the cover and all working parts are within reach.

Write for literature today.

WM. S. HAINES & CO.
460 No. 12th Street
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Here's why—

GRID UNIT HEATERS WILL LAST AS LONG AS THE PIPES FURNISHING STEAM TO THEM ----

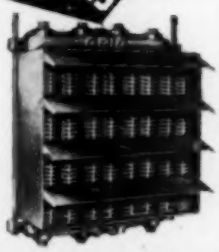
★ One piece construction heating sections (patented) of high test cast iron that withstand steam pressures up to 250 lbs.

★ No soldered, brazed, welded or expanded connections to become loose or develop leaks.

★ No electrolysis to cause corrosion, breakdowns, leaks or heating failures.

HIGH TEST CAST IRON HEATING SECTIONS

It's the material that makes the difference. GRID High Test Cast Iron Unit Heaters will withstand corrosive atmospheres, and operate efficiently under all atmospheric conditions without maintenance. In many plants GRID installations have been operating continuously for 16 years without maintenance cost. Install unit heaters that will last as long as the pipes furnishing steam to them.



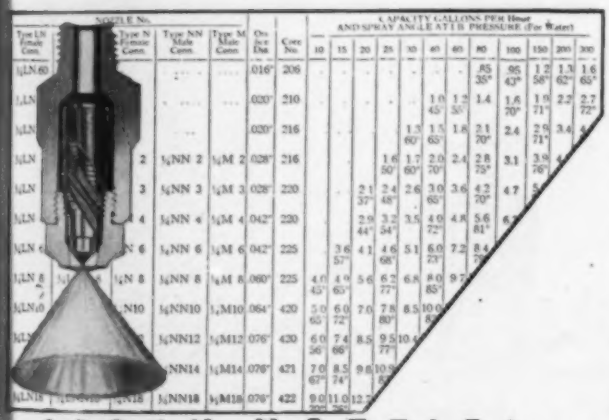
SEND for booklet "Corrosion in Unit Heaters"—free upon request.

D. J. MURRAY MANUFACTURING CO.
WAUSAU WISCONSIN

The Pipe Joint Cement that never hardens

Do the job with Rutland Pipe Joint Cement, and pipes are always easy to disjoin. Rutland seals joints tightly and lasts indefinitely, too. Saves money because more bulk per pound. Remains soft in can. Does not stain hands or fixtures.

Mail post card for free sample. Rutland Fire Clay Co., Dept. H-35, Rutland, Vt. Also makers of Rutland Retort Cement, Furnace Cement, Asphalt Paint, Concrete Patcher.



Type LN Male Conn.	Nozzle No. Type N Female Conn.	Type NN Male Conn.	Type M Male Conn.	Orifice Dia.	Conn. No.	Capacity Gallons per Hour and Spray Angle at 15 Pounds Pressure (For Water) 10	15	20	25	30	40	60	80	100	150	200	300
⅛LN 60				.016"	205								.85 35°	.95 42°	1.2 56°	1.3 62°	1.6 65°
¼LN				.020"	210						1.0 42°	1.2 50°	1.4	1.6 70°	1.9 71°	2.2	2.7 72°
¼LN				.020"	216					1.3 60°	1.5 65°	1.8	2.1 70°	2.4	2.9 71°	3.4	
¼LN	2	¼NN 2	⅛M 2	.028"	216				1.6 50°	1.7 60°	2.0 70°	2.4	2.8 70°	3.1	3.9 76°		
¼LN	3	¼NN 3	⅛M 3	.028"	220			2.1 37°	2.4 48°	2.6	3.0 65°	3.6	4.2 70°	4.7			
¼LN	4	¼NN 4	⅛M 4	.042"	220			2.9 44°	3.2 54°	3.5	4.0 72°	4.8	5.6 81°	6.3			
¼LN 6	N 6	¼NN 6	⅛M 6	.042"	225		3.6 57°	4.1	4.6 68°	5.1	6.0 23°	7.2	8.4				
¼LN 8	¼	¼N 8	¼M 8	.060"	225	4.0 45°	4.9 65°	5.6	6.2 77°	6.8	8.0 85°	9.7					
¼LN 10	N10	¼NN10	¼M10	.064"	420	5.0 55°	6.0 72°	7.0	7.8 80°	8.5	10.0						
¼LN		¼NN12	¼M12	.076"	420	6.0 56°	7.4 66°	8.5	9.5 77°	10.3							
¼LN		NN14	¼M14	.076"	421	7.0 62°	8.5 74°	9.8	10.9								
¼LN18	¼NN18		¼M18	.076"	422	9.0	11.0	12.5									

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Whatever your requirements may be, Spraying Systems Spray Nozzles offer two important advantages: (1) spray nozzle design specific to every basic application and (2) a wide range of sizes in each type, for exact performance to fit the job. Catalog No. 22 will give you all the facts.

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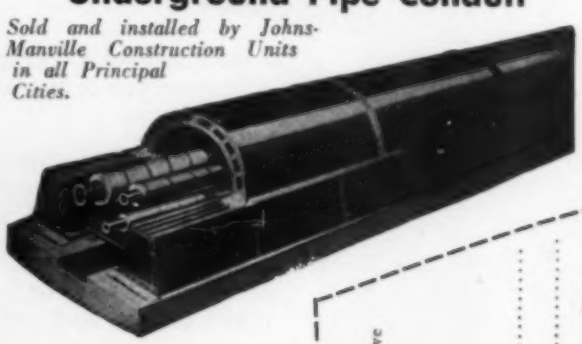

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☐ Please send representative.
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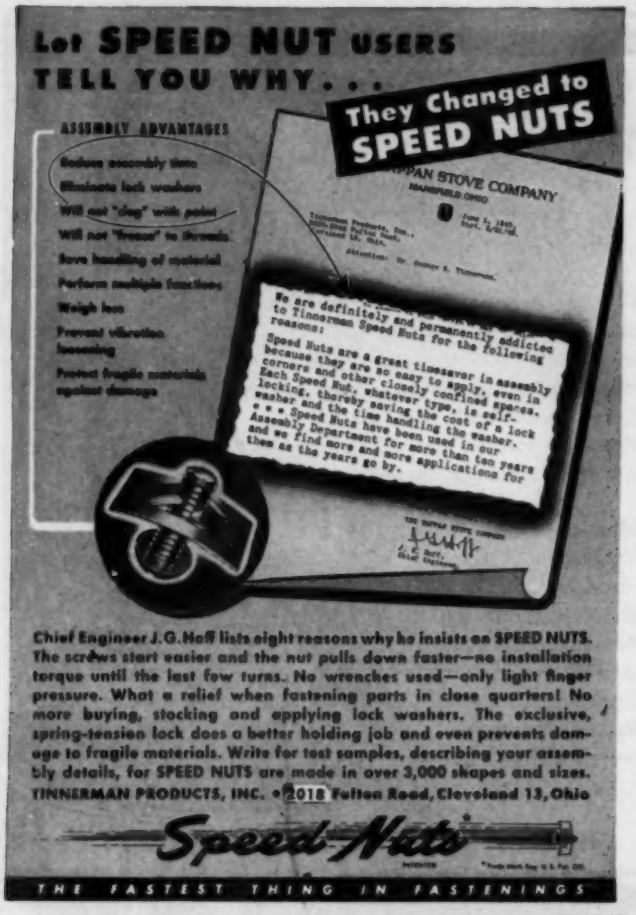
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The chart may be used to determine the tempers available, ranges of tensile strength, yield strength, hardness, elongation, and comparable AMS, ASTM, Federal, and Navy specification for the products offered by the manufacturer. These aluminum alloy products are segregated on the chart in seven groups; i.e., sheet, plate, wire, rod-bar, shapes, tubing-piping, and forgings.

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Charts are available from Reynolds Metal Co., Dept. 47, 2500 S. 3rd St., Louisville 1, Ky. The price is $1.00 each.

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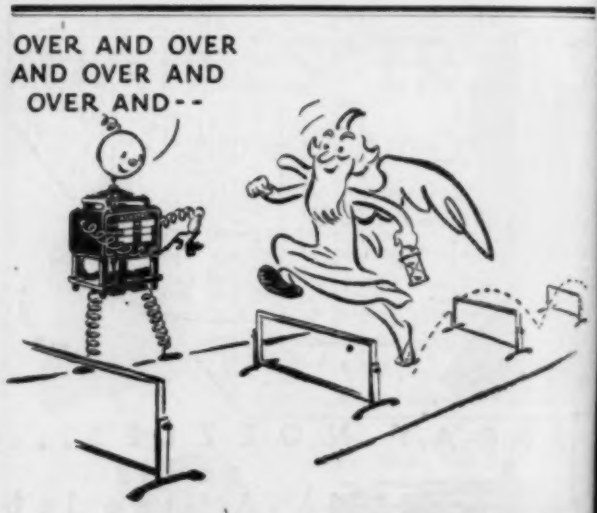

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